18 Industrial and Marine Engines

Günter Kampichler, Heiner Bülte, Franz Koch, and Klaus Heim

18.1 Small Single Cylinder Diesel Engines

18.1.1 Introduction

The industrial single cylinder diesel engine has a history steeped in tradition, which extends from the first operable diesel engine in 1897 up to today's versatile, air-cooled, small single cylinder diesel engine. On account of the low manufacturing costs relative to output, lowfuel consumption, good lubrication conditions and better exhaust quality, it is now only built as a four-stroke engine, primarily in the small diesel engine segment.

Despite the high level of development achieved, potentials for improvement continue to be researched. These are chiefly envisioned in the use of new, high grade materials and in the criteria of air/fuel mixture formation and control.

While the service life and reliability of industrial engines will always have top priority, complicated market mechanisms as well as additional parameters, e.g. exhaust and noise emission, increasingly demand consideration (see Chaps. 15 and 16).

Technical advances in related types of product ranges are also spurring on the development of small diesel engines. However, the same standard cannot always be applied: the development of a new engine generation or the introduction of new technologies necessitates meticulous studies, including exact analysis of present engine engineering, which has proven itself in harsh industrial operation for decades. Only a correct assessment of foreseeable stages of development and future requirements will make it possible to integrate new technologies in internal product development at an early stage and successfully market these products with an edge over the competition. Fields of application for engines with a power range of 2–12 kW at 3,000 to a maximum of 3,600 rpm are construction equipment, municipal vehicles, lawn and garden and agricultural equipment and small tractors, electric generators, water pumps and boat engines.

The classic small single cylinder diesel engine. Horizontal designs are only still found in Asia. Typically implemented as gasoline lawnmower and outboard engines earlier, vertical shaft engines also became available as single cylinder four-stroke diesel engines four years ago (Fig. 18-1). Free inertial forces and moments of inertia are less noticeable in devices with a low location of center of gravity, e.g. lawnmowers.

Engine customers' demands are diverse and extremely varied. They all want to see their current case of installation optimally resolved in terms of function and cost. Responding to every customer demand would result in a multiplicity of different engines. This is only worthwhile when backed up by suitable quantities, which is normally not the case. Consequently, all engine manufacturers attempt to develop their own concept strategies, which, taking a basic engine as the point of departure, can provide maximally variable, universally implementable and customizable engines. This presupposes installation engineering in a dialog between engine suppliers and users. Harmonization of requirements specifications and consultation supporting installation up through the start of a device's production and including acceptance certification have become part of engine manufacturers' standard programs.

18.1.2 Performance Specifications and Basic Data of Single Cylinder Diesel Engines

18.1.2.1 Power Range and Combustion Process

Engine Power

The market demands four-stroke single cylinder diesel engines with powers of 2–12 kW and cylinder volumes of V_h = 200–850 cm^3. Annual worldwide demand for such engines is 1.2 million and growing. Two cylinder engines already make sense above 12 kW of power. Allowing for costs, a lower limit of 8 kW at 3,600 rpm is justifiable depending on the torque requirement. Low free inertial forces and thus gentler, lower vibration engine operation are crucial. Using higher speeds to obtain a power increase is detrimental to the application and hardly effective. Maximum rated speeds of 3,000 rpm in the upper displacement range or up to 3,600 rpm in the lower displacement range have proven themselves.

K. Heim (✉)
Wärtsilä Switzerland Ltd., Winterthur, Switzerland
e-mail: klaus.heim@wartsila.com

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Combustion Process

The basic requirement for single cylinder diesel engines continues to be manual startability down to at least –6°C (–12°C) without electric auxiliary devices such as glow plugs or coils. Since a swirl chamber engine is unable to start below 0°C without preglow, direct injection (DI) is the only option for small displacements (≤0.4 l).

18.1.2.2 Engineering Requirements

Stroke to bore ratio. While a stroke to bore ratio of s/D>1 is fundamentally preferable, larger strokes impede exploiting the advantages of optimal overall height. Occasionally, this may be the decisive factor for the implementation of a motorization project. Thus, even engines with an s/D ratio of up to ≥0.6 are available. Vertical shaft engines are gaining popularity, e.g. for installation in lawnmowers, because of their overall height (see Fig. 18-1).

Cooling systems. For all intents and purposes, only direct air cooling (see Sect. 9.1.4) with a radial exhaust fan integrated in the flywheel is employed. This space saving and low cost principle utilizes baffles to systematically route cooling air to temperature-critical components such as the cylinder head and cylinder liner. The larger the engine surface area drawn on for cooling, the larger the reserve is for implementation in countries with high ambient temperatures. Water cooling of the type in small multiple cylinder engines is too complex and hence out of the question.

Continually tightened regulations for noise emission of appliances, equipment, etc., also target the drive, i.e. the combustion engine. Thus, full engine encapsulation is often the only corrective (see Sect. 16.5). The thermal problems that arise in an engine and the required damping of cooling air noises are compelling engine manufacturers to develop new concepts for fully enclosed engines. Hatz is testing liquid cooling with lubricating oil and an external heat exchanger for its B model engines (Fig. 18-2) with a modified cylinder head, oil circulation on the control side and a larger oil supply. The enlarged, oil-filled gap between the slip-fit cylinder liners serves to cool them or acts as a heat bridge to the housing.

Engine mount. When possible, an engine should be mounted elastically. The supplier industry provides

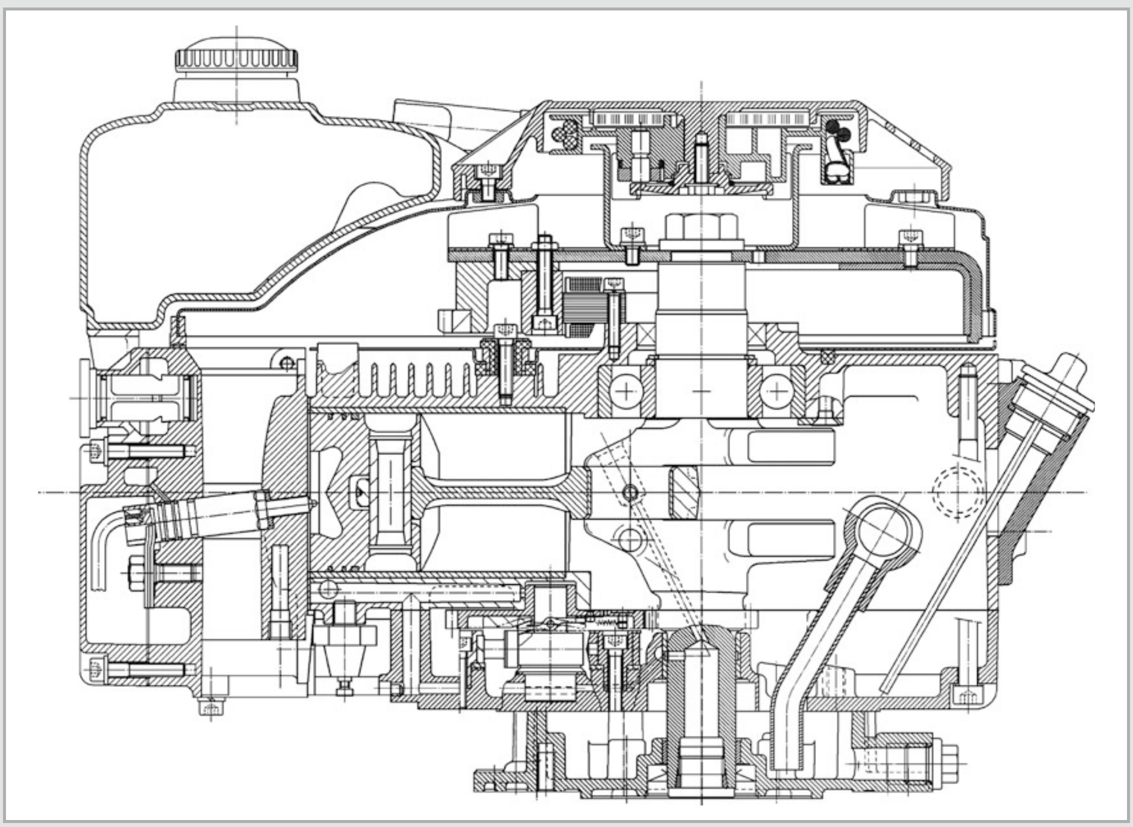

Fig. 18-1 1B20V vertical crankshaft engine $V_h = 232$ cm^3, $P_e = 3.8$ kW/3,600 rpm

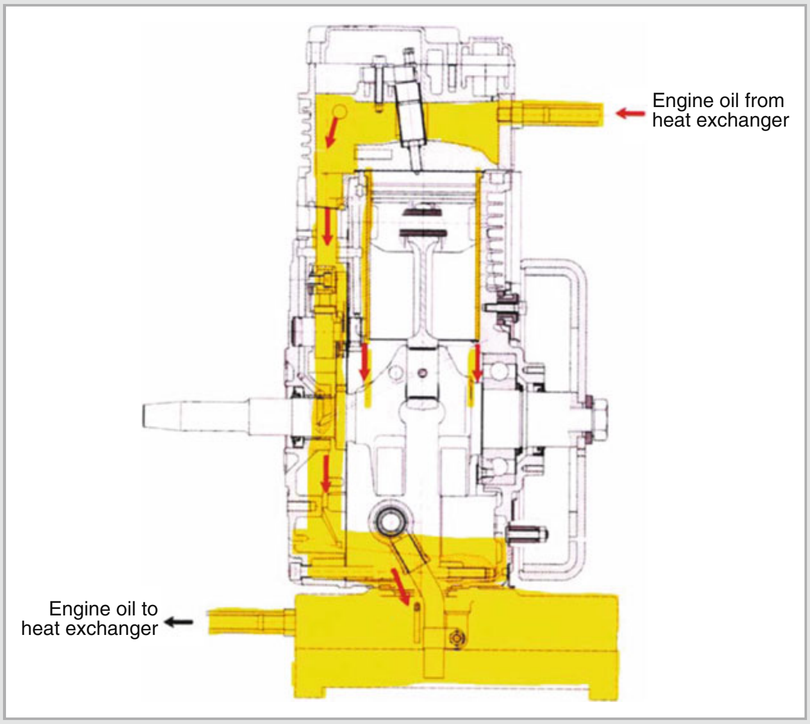

Fig. 18-2
Liquid cooling with engine oil as cooling medium and an external heat exchanger in Hatz B series (in testing)

extensive options. Rubber/polymer elements with hydro mounts have proven to be excellent even when resonance speeds are run through when an engine is revving or coasting when it is stopped.

Belt driven machines may initially be rigidly constructed on an intermediate frame together with the engine. Damping elements for the foundation can decouple the frame. An engine may be designed rigidly in conjunction with very stiff and massive frames and foundations.

Power Take-offs. Universal implementation of an engine requires output on both the flywheel side and the opposite control side. This reverses the direction of rotation. However, the recoil starters increasingly being implemented block power take-off on the flywheel side. Depending on the engine design, dispensing with the simple manual starter and selecting the more convenient but also more vulnerable electric starter produces other power take-offs. For instance, the 1D81 engine has 4 PTOs (see Fig. 18-3).

In addition to the options of counterclockwise (standard) or clockwise rotation, torque may also be decreased:

- 100% axially on the flywheel side by a coupling flange or radially by a belt pulley (1),
- 100% radially and axially on the control side at the crankshaft (2),
- additionally 100% on the control side when hand crank started and at the camshaft by attaching a V-belt pulley (3) and
- to a limited extent at the camshaft that drives small hydraulic pumps (4).

Starter Systems, Starting Option

Hand crank start. A hand crank start always requires a higher mass moment of inertia than an electric start. However, a higher mass moment of inertia means considerably higher engine weight because of the necessarily heavier flywheel. Potential corrosion in electric components and damage to batteries by strong shaking are points against electric starting in harsh construction equipment.

Neglected maintenance, improper operation, etc. typify the construction equipment rental business where “simple”, “rugged” and “functionally manageable” are demanded. These are arguments for a hand crank start in larger single cylinder diesel engines upwards of $V_h > 0.5$ dm^3 in which several preselectable, decompressed cycles speed the flywheel up enough that its energy allows engine auto-ignition and revving for one or more TDC cycles at full compression. German laws mandate protective measures against dangerous recoil (reversion) when the flywheel action is inadequate and

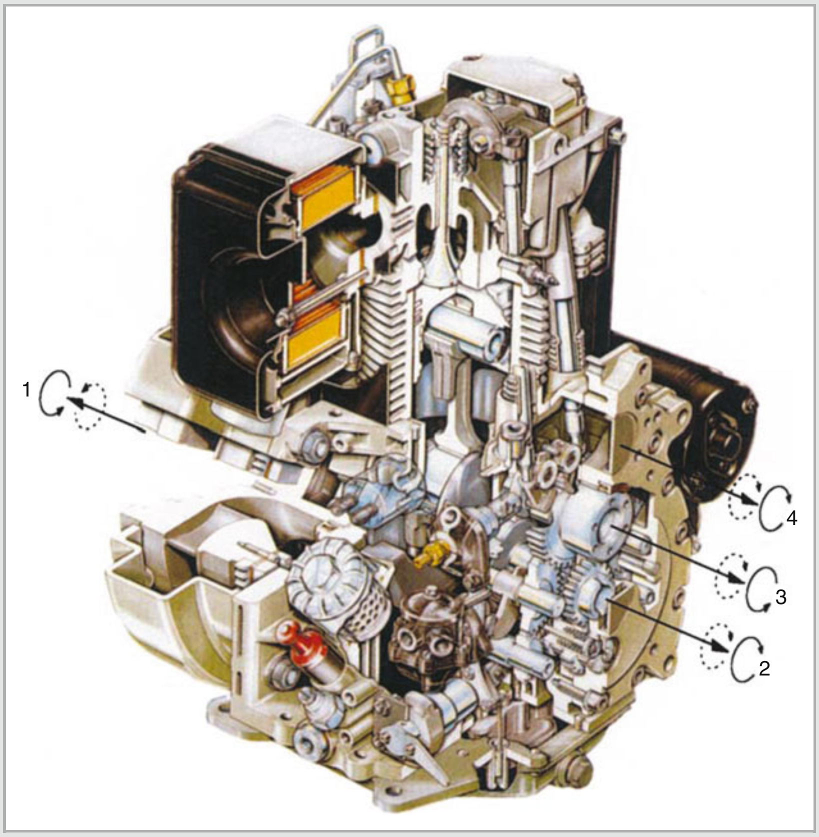

Fig. 18-3
PTOs on a single cylinder air-cooled diesel engine (HATZ SUPRA 1D81). $V_h = 0.667\ dm^3$

ignition starts before TDC, e.g. a trip gear installed in the hand crank that stops the transmission of power to the recoil torque after a few angular degrees. Hand crank starting is feasible up to a displacement of 0.8 l when the flywheel and the multiplication of hand crank to crankshaft speed are selected correctly. A hand crank speed of 2.5 revolutions per second with a maximum hand crank radius of 200 mm at full expenditure of energy over a time of 3.5 seconds is barely justifiable ergonomically.

The trend toward using flywheels to decrease engine mass related to power comes at the expense of dependable starting and cold start performance. Figure 18.4 presents the correlations for dependable cold start at –6 °C.

Recoil start. Implemented in stationary *gasoline* engines virtually without exception, recoil starters are increasingly also being employed in *diesel engines*, especially since they present hardly any danger of injury. The energy manually introduced by a cable with a deflection of 0.7–1 m must bring the flywheel to the starting speed within one cycle, i.e. two revolutions. Starting is facilitated by decompression by elevating the exhaust valve by approximately 0.1–0.2 mm by a simple lever system that returns after one cycle or by an automatic centrifugally controlled decompression system. Figure 18-4 also includes the moment of inertia of the flywheel for flicker-free generator operation at 3,000 min^1. However, this already makes cold start by means of recoil start problematic as of a displacement of 300 cm^3 and above. Superior in this case, hand crank start allows higher transmission ratios, which are between 4:1 and 5:1 in IDI engines with $V_h < 0.4$ l. Prechamber engines (IDI) additionally require electric preglow units below –6°C.

Electric start. With the exception of the harsh construction sector, starting with electric starting motors using pinions and a ring gear at the flywheel is increasingly being implemented in air conditioners and elevating and lifting equipment, for example, for which remote control or control electronics have been designed in from the outset. Since single cylinder diesel engines have long settling times (up to 2.5 s), a tooth modulus >2.5 mm has to be selected that prevents teeth from breaking out during inadvertent post-starting. Hence, an electronic start block relay, which is indispensible for remotely controlled engines, is generally recommended. Germany and Europe will continue to prohibit "ignition aids" sprayed into the intake tract during manual starting as well as simple

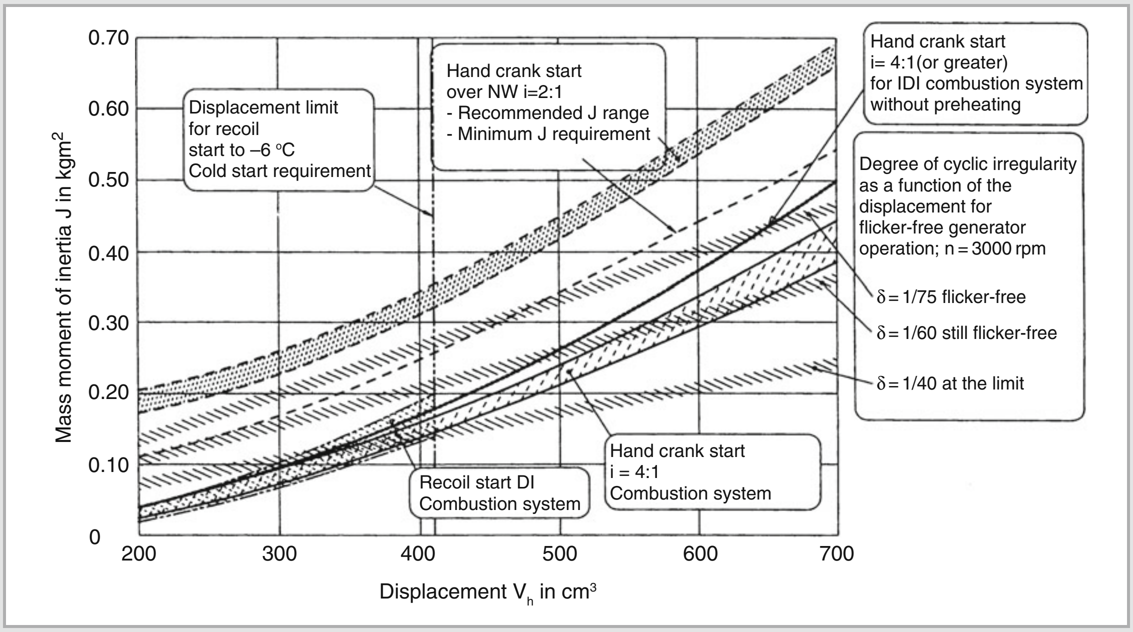

Fig. 18-4 Flywheel moment of inertia as a function of displacements required for single cylinder diesel engines at the cold start limit of –6°C

manual pull starters in the future because of the danger of accidents.

Intake and Exhaust Systems

Air Cleaners. Formerly used for small diesel engines, oil wetted air bath cleaners have poorer filtration efficiencies but are easy to service since they function with the engine oil available at construction sites. Significantly better in terms of their filtration efficiency, dry air cleaners (see Sect. 13.1) not only require the stocking of cleaner cartridges but also maintenance during which cleaner loading is checked. The common cleaner vacuum indicator has problems in single cylinder engines because of the strong intake pulsations. A clogged cleaner cartridge can already initiate severe damage in the cylinder head region after a brief time by overheating due to a deficiency of combustion air. Since increased dust concentrations occur in the construction sector, a basic engine concept should design in both types of cleaners and allow the installation of a precleaner (cyclone) when dust concentrations are extreme. Oil wetted air bath cleaners are preferable when the supply of replacement parts is uncertain, e.g. in "third world countries".

Intake tract. An airtight connection, e.g. elastic hoses, has to be provided from the air cleaner's raw air inlet up to the engine's intake air inlet. When engines are installed in semi or completely enclosed spaces, the air should be supplied externally without pressure losses and temperature increase (recommended values: $\leq$10 mbar (100 mm WS) vacuum and 5 K temperature increase over the external air, measured at the intake port inlet at rated speed).

Exhaust system. The maximum muffler volume should be selected. On the one hand, structural space renders a volume that is ten times the displacement unfeasible even though this is ideal in terms of power loss and tailpipe noise. On the other hand, no effective damping without power losses of up to 10% is achievable below three times the displacement. The mean back pressure at the muffler inlet (= exhaust port outlet) should be <25 mbar (250 mm WS) at rated speed. Apart from being sealed tightly, exhaust manifolds have to be installed rigidly or with flexible compensators depending on the engine mount to decouple vibrations. Engines installed inside enclosed spaces require the shortest insulated exhaust manifolds possible.

Engine compartment ventilation. When engines are installed in tight, enclosed spaces (encapsulation), the cooling air has to be supplied to the impeller inlet with a minimal temperature increase (<3 K). Bellows, hoses or ducts should discharge heated exhaust air to the atmosphere by the shortest path to regulate the heat balance in the engine or installation space. Partial evacuation from the installation space by means of the flywheel fan is normally sufficient when the muffler is mounted outside the installation space and the exhaust manifold is short and insulated. When a muffler is located in an engine compartment, it as well as the following exhaust pipes must be placed in a compartmentalized air duct. External ventilation is only necessary when the exhaust flow is not leak proof.

Fuel Supply

A fuel tank mounted at the engine with a filter, a fuel line to the injection pump and return and leak oil return from the nozzle holder to the tank are standard. Reliable engine operation requires a minimal gradient of $\geq$50 mm from the tank to the injection pump, allowing for potential operation in an inclined position. Horizontal or slightly inclined piping arrangements of $<10^\circ$ have to be avoided as well. This is particularly true for the forward flow from the tank to the injection pump. The discharge of gas or air bubbles in the fuel after engine shut off has to be assured, particularly in the region of the injection pump. When fuel is supplied by a fuel tank positioned lower, a non-return valve installed directly before the diaphragm feed pump prevents the fuel system from discharging when the engine is stopped. Otherwise, protracted ventilation procedures would be required during restarting.

Alternators

Alternators in single cylinder diesel engines are usually designed as space saving flywheel alternators. Magnet segments arranged in a ring on the flywheel bypass star-shaped spool bodies attached to the crankcase with a clearance of approximately 0.4 mm. The voltage regulator is attached to the engine so that it is easily accessible and delivers a rectified charging current of 15 A at 12 V and 8 A at 24 V as a function of speed and thus a charging power of $\approx$200 W. In another concept, permanent magnets mounted on the flywheel interact with a coil bracket attached to the crankcase through an axial air gap. It alone is replaced when there is a malfunction. Thus, the engine or flywheel does not have to be removed. Alternator power can be boosted easily by additional coils.

18.1.3 Engineering Design of Small Single Cylinder Diesel Engines

18.1.3.1 Crankcase

While gray cast iron crankcases provide advantages in terms of noise emission, they are no longer relevant because of their weight just as lightweight sand or gravity die cast aluminum crankcases are no longer relevant because of the expensive machining (Fig. 18-5).

Crankcase or rack designs that can be cast from light alloys are primarily employed for reasons of cost. One technically optimal, low cost solution is a pressure die cast aluminum crankcase design with an integrated, raised holder for the cylinder liner, which is open on one side and thus easy to demold (see Figs. 18-6 and 18-7). As the sealing component, the control cover accommodates one main bearing. However, the development of the force lines reveals the limits of this design principle. Firing pressures that are no longer manageable are produced at displacements that are larger than 0.6 dm^3. Here, a one-piece design open on the bottom, which can be pressure die cast, in

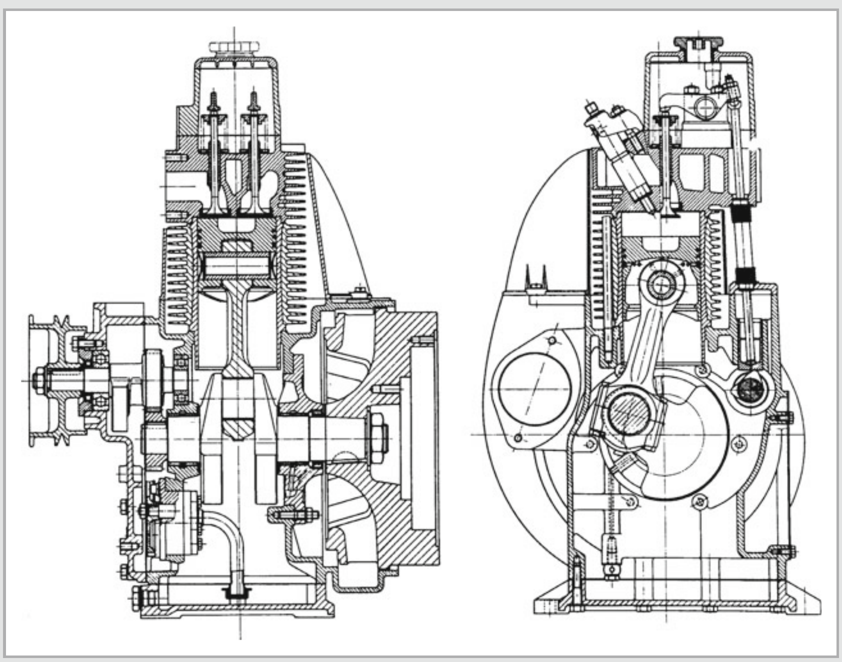

Fig. 18-5
Older model single cylinder air-cooled diesel engine with gray cast iron crankcase. $s/D = 82/82$ (DEUTZ F1L 208)

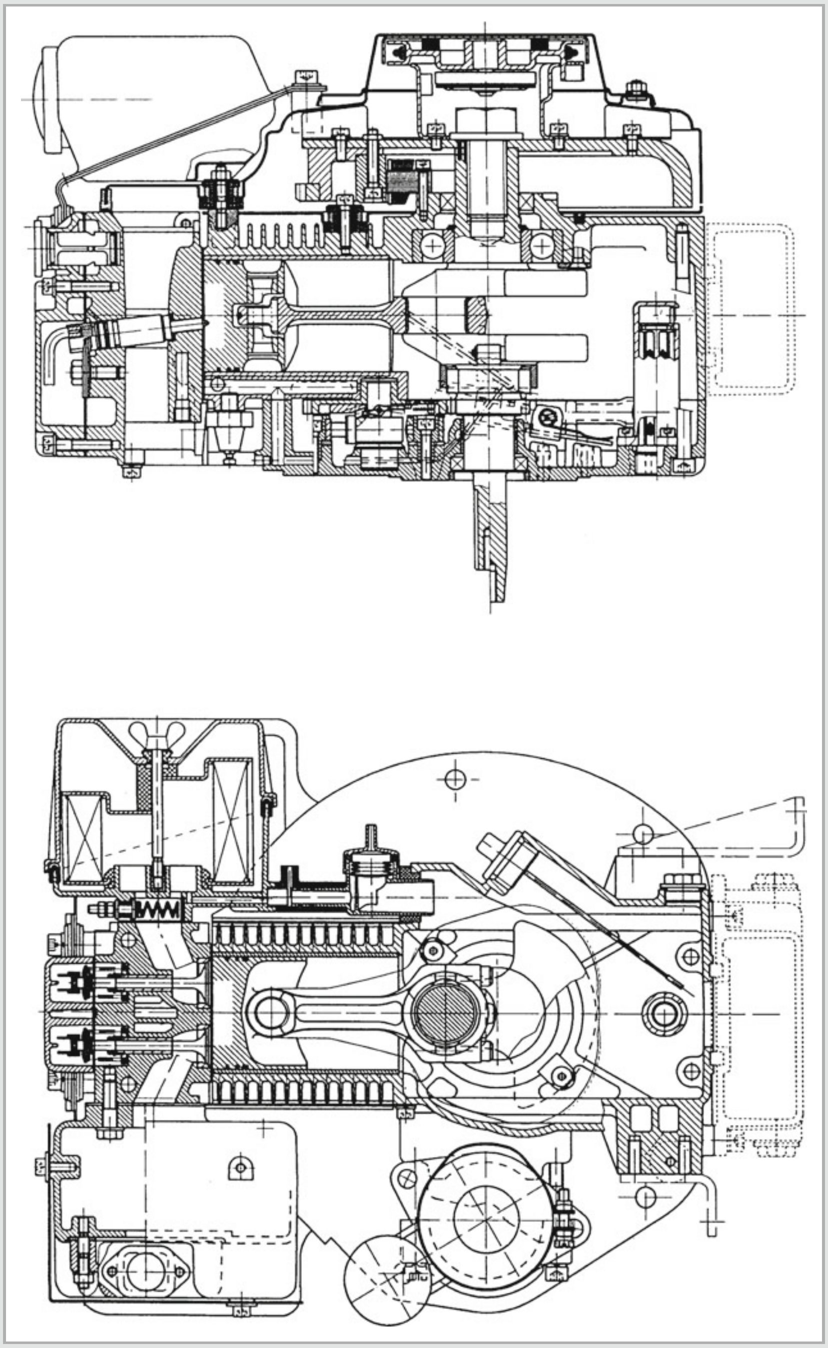

Fig. 18-6
Air-cooled single cylinder light duty diesel engine with a pressure die cast light alloy crankcase in "one side open" design, direct injection system and single cam system (SCS) (HATZ 1B20). $V_h = 0.23\ dm^3$; $s/D = 62/69$

which a bolted-on oil pan seals the case as in multiple cylinder engines, is suitable for larger single cylinder diesel engines.

Camshaft and governor. The camshaft may be positioned in a variety of ways. It may be arranged laterally parallel to the crankshaft as in inline engines (Fig. 18-7), centered above the crankshaft or, optionally, on the control or flywheel side (Fig. 18-6). The transmission element of the P-governors utilized to control injection pumps are usually driven by the camshaft or oil pump gear. Stepped

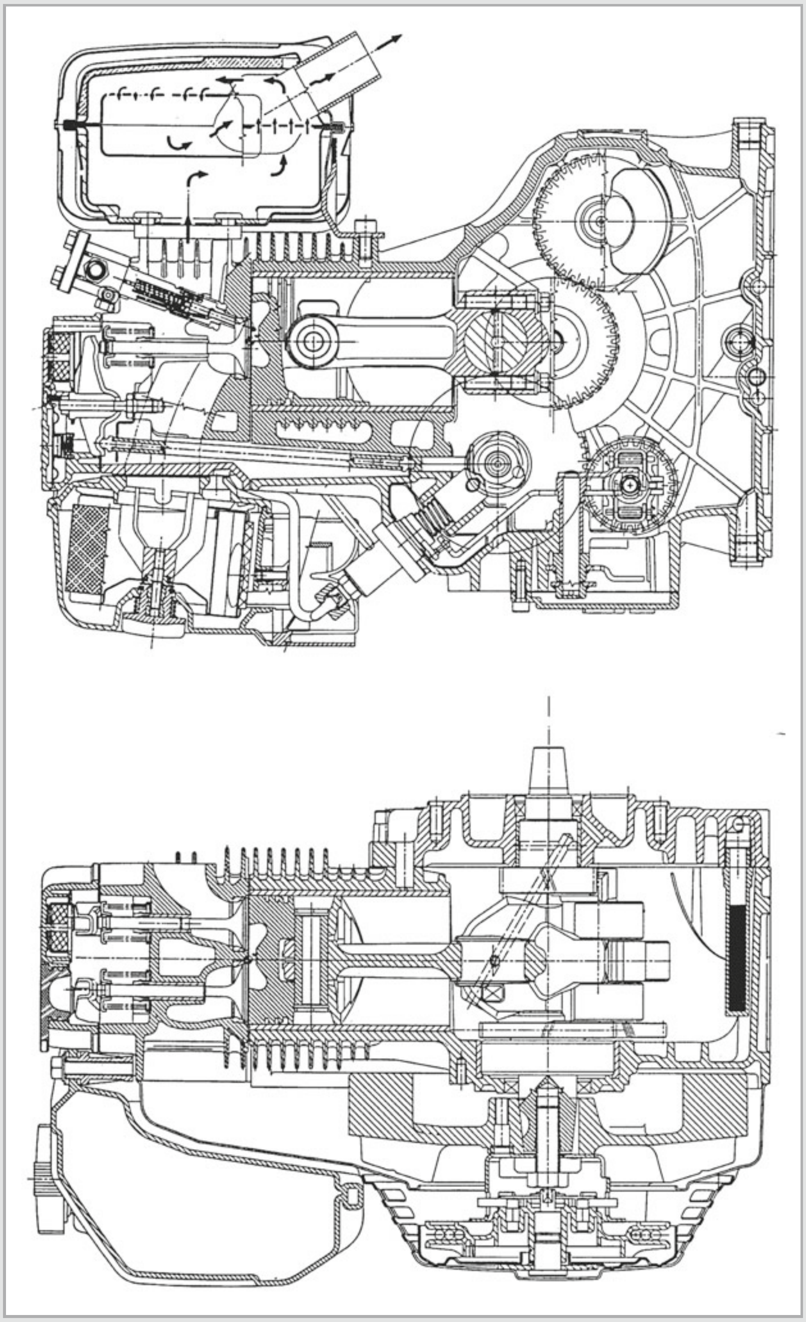

Fig. 18-7
Air-cooled single cylinder light duty diesel engine with pressure die cast light alloy crankcase in "one side open" design and direct injection system (LOMBARDINI 15LD315). $V_h = 0.315$ dm^3; s/D = 66/78

up transmission ratios are advantageous for control and reduce the space required for a governor. Every engine manufacturer has its own systems, which are more or less flexible depending on the use. A variable-speed governor, ± torque control and a freely selectable proportional degree of 3–10% (3–5% for electric generators) are standard. Camshafts with injection cams and valve gear assemblies are usually housed in the open space between the control cover and crankshaft web together with the injection pump, speed governor and oil pump drive (see

Fig. 18-7). Accepting a free moment of inertia, only the balance shaft for 100% first order mass balancing is located on the flywheel side.

Crankshaft Assembly

Crankshaft. A forged crankshaft is quenched and tempered and hardened in the region of the main bearing journal and the crank pin. Plain bearings still stand for long service life, roller bearings for low friction. Recently, PTFE coated plain bearings have also been being used to reduce friction, above all cold friction. Deep-groove ball bearings now reach calculated expected service lives of 4,000–5,000 hours of operation, which is sufficient for standard single cylinder diesel engine applications. Roller and plain bearings are often combined when the roller bearing is mounted on the flywheel side. Nodular cast iron crankshafts are conceivable for single cylinder engines but still remain to be proven in use in rough construction equipment. Composite crankshafts are too unreliable even in micro diesel engines and remain reserved for gasoline engines.

Connecting rod. A standard connecting rod is forged of steel, split in the connecting rod big end and given a bronze bushing in the piston pin boss. When they are appropriately designed, forged aluminum connecting rods may be implemented in short-stroke engines with displacements of less than 0.3 l. Connecting rods made of sintered material may be expected in the future. GGG 60 may also be used as a connecting rod material with the advantage of operation without a bushing in the small piston pin boss.

Pistons. Only aluminum alloy full skirt pistons are employed because of their minimum of oscillating masses (see Sect. 8.6). Control pistons are the exception and only expedient in encapsulated single cylinder engines.

A standard assembly consists of three rings:

– a usually chromium plated compression ring that is a rectangular or keystone ring, convexly finish machined and tapered,
– another tapered compression ring that is a Napier ring or a ring with an inner bevel and
– an oil scraper ring that is a top beveled or double-beveled ring with or without spring support (spiral expander ring).

Flywheel. Flywheels predominantly consist of gray cast iron with cast-in blading. Plastic fan rings may also be bolted onto flywheels. Consisting of several layers of plate and thus anechoic, deep drawn steel flywheels are the trend in small engines (see Fig. 18-6).

Mass balancing. Comprising 35–70% of the oscillating masses in addition to the rotating mass fraction, bolted-on counterweights are common for mass balancing (see Sects. 8.1 and 8.3). Fifty percent of first order mass balancing (normal balancing) usually suffices below a displacement of 0.5 l. One hundred percent of mass balancing of first order inertial forces is frequently connected with an additional free moment (see Sect. 8.3). The engine pictured in Fig. 18-3 (HATZ SUPRA series 1D30/40/60/80) achieves comparatively ideal 100% first order mass balancing. Figure 18-8 illustrates the principle: A single small counterweight at the flywheel side crankshaft web makes small overall engine height possible and simultaneously frees space on the control side to operate a balance shaft system that rotates counter to the direction of rotation in an almost ideal position (see Sect. 8.3.6). All other counterweights are arranged in the flywheel without interfering inside the housing.

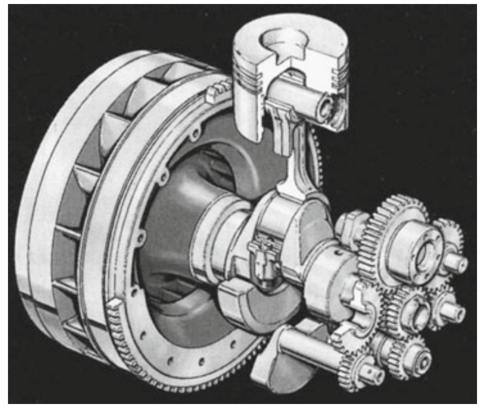

Fig. 18-8 Arrangement of balancing masses and balance shafts to completely balance first order inertial forces (HATZ-SUPRA concept line; see Fig. 18-3)

Cylinder Liner, Cylinder Head and Valve Gear Assembly

Cylinders. Finned, compact gray cast iron cylinders attached to the crankcase with a continuous tensioning bolt design are employed because of their dimensional stability. Fitted aluminum finned cylinders with cast-in gray cast iron liners are also widespread.

Crankcases with integrated cast-in gray cast iron cylinder liners raised up to the top face of the cylinder head are also state-of-the-art for engines with $V_h < 0.4$ l and preferred for reasons of cost (see Fig. 18-7). In its B engine series, Hatz employs a centrifugally cast liner floating on an oil cushion. Its advantage is the option of replacement during maintenance (see Fig. 18-6). Single cylinder air-cooled diesel engines predominantly have cast aluminum alloy cylinder heads with spiral swirl ports inserted in the Croning shell molding to generate air swirl for mixture formation during direct injection (DI). The methods of core splitting and extraction suited for pressure casting employed by Hatz for the cylinder heads of its B series provide a very interesting solution (Fig. 18-6). The valves are solely actuated by rocker arms and push rods, either the

cam followers or the interconnected drag levers having direct contact with the cam. Forged or deep drawn sheet metal arms are employed as rocker arms. Hatz has effectively implemented its patented purely mechanical valve clearance compensation (see Fig. 18-9) together with rocker arms fabricated from sheet metal in its B series engines. The intake and exhaust valve and the injection pump drive in these engines are actuated by only one cam profile path (the patented single cam system SCS in Fig. 18-10), thus saving space.

Injection system. The PLN (pump-line-nozzle) system is dominant. Direct injection (DI) requires short injection lines to minimize dead space volumes in the high pressure system and maximize the drive's dynamic rigidity. Driven by roller tappets or overhead camshafts, the UPS (unit pump system) particularly meets these requirements and, hence, is considered an alternative for future small single cylinder diesel engines in terms of installation space too.

Shaft nozzle holders and P type nozzles with needle diameters of 4 mm and thus less moving mass are standard for PLN systems. Dual spring nozzle holders allow pilot injection that reduces combustion noise at light load but with the drawback of high smoke values in the full load range. Multijet injection would also be desirable for small diesel engines but must be ruled out because of the high costs. RSN nozzles have proven to be very suitable. They attain variable choke control at a larger needle stroke and thus reduce both combustion noise and NOx and CO emissions. Even though they are suboptimal, simple hydraulic measures have to be relied on in the pump element and in the injection valve since electronic components that control the start of injection either have to be ruled out for reasons of cost or are unavailable for small single cylinder diesel engines.

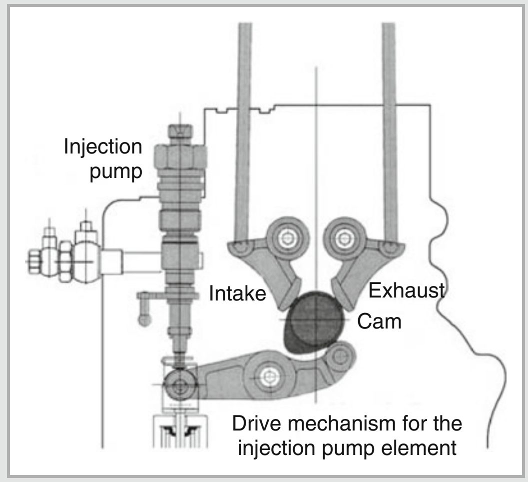

Fig. 18-10 Schematic of the patented single cam system SCS (HATZ)

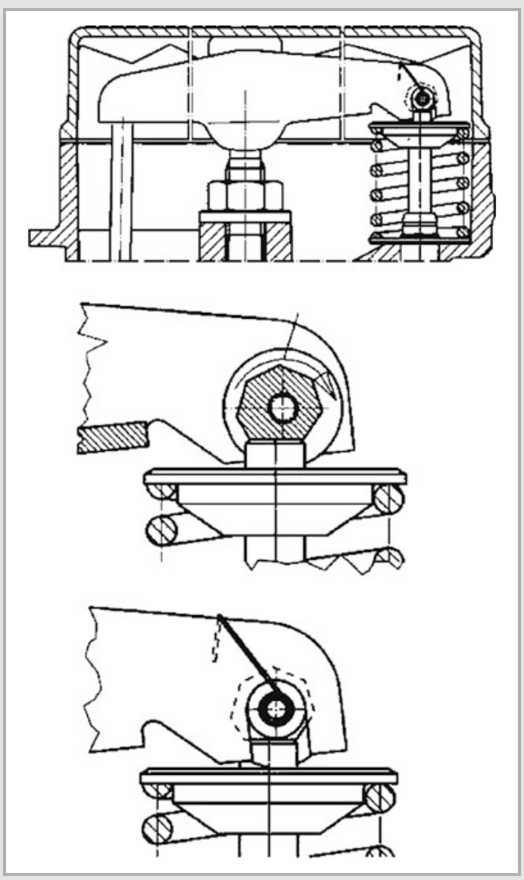
Fig. 18-9 Mechanical valve clearance compensation with an automatic eccentric multistep catch (torsion spring driven)

18.2 Stationary and Industrial Engines

18.2.1 Definition and Classification

The term stationary and industrial engines refers to nearly every combustion engine that has been modified and certified for use outside of road traffic, so-called nonroad applications. While vehicles with industrial engines may be operated partly on public roads, e.g. tractors, street cleaners or front loaders, the proportion of road traffic is only of secondary importance for these applications. They are certified in a cycle of levels based on the application in compliance with ISO 8178 (see Sect. 15.2). Typically, stationary and industrial engines are often implemented in a multitude of applications in very low quantities. This compels manufacturers to produce engines in a modular design in order to satisfy the conditions of a particular installation. A modular system that covers a maximum number of potential applications without allowing the

diversity of variants to grow to uneconomical scales is essential for add-on parts. The applications can be roughly classified in three groups:
– stationary engines,
– mobile machinery and
– agricultural equipment.

Stationary engines basically serve to generate power (generating sets) but are also employed for other units, e.g. cooling systems, pumps and compressors. Depending on the application, they are in continuous operation that may largely consist of light load operation or intensely intermittent operation followed by high load, e.g. emergency power units. Stationary engines are operated with variable loads but at constant speed. This particularly pertains to power units, which are operated in Europe at 1,500 rpm to guarantee a constant alternating current frequency of 60 Hz and in the USA at 1,800 rpm to guarantee 50 Hz. They are certified in the D2 cycle in compliance with ISO 8178 (see Sect. 15.2).

The domain of mobile machinery applications covers the large sector of construction equipment such as excavators, front loaders and bulldozers as well as forklifts, rail vehicles and airfield tractors. Depending on the application, engines in mobile machinery are operated in the entire map as well as at a fixed operating speed when the performance requirement is controlled by a hydraulic unit. The operating speed usually corresponds to the engine's rated speed. Train engines are either certified like industrial engines or in accordance with emission legislation for on-road commercial vehicle engines.

Agricultural equipment actually corresponds to mobile machinery in its applications but constitutes a distinctive category in terms of performance requirements and installed components. Therefore, agricultural equipment engines also constitute their own group technologically. Frequently, a distinctive design feature of agricultural equipment engines is their assumption of a stiffening function of the vehicle.

Based on their design, industrial engines may be classified as modified vehicle engines and as engines systematically developed for industrial application. Modified vehicle engines include both modified car engines for ratings of up to approximately 100 kW and commercial vehicle engines for ratings of up to approximately 500 kW. Industrial engines generally cover a power range from 2 to approximately 500 kW. Applications above 1,000 kW are operated by medium speed and large low speed engines (see Sects. 18.3 and 18.4). Only very few engines are in the power range between 500 and 1,000 kW.

Just as for car and commercial vehicle engines, electronically controlled injection systems are also becoming established for industrial engines, above all for engine powers above 75 kW. In effect since 2006, level 3 emission control legislation for industrial engines has made it impossible to meet the requirements in this power range with mechanically controlled injection systems at a justifiable cost. Unit pumps, unit injectors and, increasingly, common rails vie with one another as injection systems. Moreover, the electronics also allows the integration of customer functions and intelligent linkage of engine electronics with vehicle or equipment electronics (see Chaps. 5 and 6).

Many engines below 75 kW are equipped with mechanically controlled injection systems because the emission requirements are less stringent. Since an engine's purchase price plays a more dominant role than operating costs in this market segment, the advantages of engine electronics are foregone.

Industrial engines were originally only constructed as air-cooled engines (see Sect. 9.1.4). Dispensing with an additional cooling medium translates into an indisputable advantage for handling and maintenance in terms of reliability, especially in sometimes extremely harsh operating conditions, e.g. under extreme climatic conditions. The gradual tightening of emission limits is causing air cooling to be edged out partly by water cooling since lower component temperatures give the latter an advantage in terms of nitrogen oxide emissions as well as power density. Contrary to original expectations, developers have always successfully complied with every limit level in the past years, even with air-cooled engines. Thus, a market for air-cooled engines may be expected to continue to exist even in the near future. The manufacturer Deutz is very successfully marketing a concept for oil-cooled engines (see Sect. 9.1.3). The basic concept very closely approximates the design of water-cooled engines but avoids the additional cooling medium of water. Since oil temperatures are generally higher than the coolant temperature of water-cooled engines, oil-cooled engines are also operated with somewhat higher component temperatures (see Sect. 9.1.3).

18.2.2 Range and Selection

Despite the concentration process witnessed in the automotive and engine industry in recent years, the range of engines worldwide remains extremely large. Thus, this can neither be treated in detail nor in full here. Table 18-1 conveys an impression of the fields of application for diesel engines worldwide. The significant applications come from the car, commercial vehicle and agricultural equipment sectors, thus documenting the small quantities the multitude of applications for stationary and industrial engines normally entail. Nevertheless, numerous well known vehicle manufacturers also offer industrial engines. Figure 18-11 provides an overview – without any claim of completeness – of the global range of industrial engines.

The use of modified vehicle engines in the industrial engine range particularly provides advantages generated by mass production. However, this is frequently accompanied by little flexibility in terms of variations in installation and flexibility is precisely what manufacturers specialized in the production of industrial engines are strong in.

A distinction is made between so-called captive and non-captive manufacturers of industrial engines. The core

Table 18-1 Use of manufactured diesel engines (100 units)

Region	Japan	East Asia	North America	Western Europe	Eastern Europe	Worldwide total
Passenger cars	323	167	0	4,383	1,013	6,209
Commercial vehicles	774	1,047	693	2,328	277	5,853
Agricultural machinery	590	7,156	42	340	67	8,792
Construction equipment	299	61	112	271	22	812
Other industrial engines	140	47	31	186	9	482
Power units	204	179	17	247	28	711
Marine engines and auxiliary marine engines	38	203	13	35	3	297
Total	2,368	8,860	908	7,790	1,1419	23,156

business of captive manufacturers is the production of industrial machinery or vehicles. They cover the range of engines necessary for this machinery with their own engine production. Some typical manufacturers in this market are Caterpillar, John Deere and Yanmar. This market segment is called captive because it is unattainable for other engine manufacturers. However, the aforementioned manufacturers also sell their engines on the non-captive market and thus compete with pure engine manufacturers such as Cummins or Deutz.

In light of the increasing complexity also driven by emission control legislation, a concentration process also observable in the automotive industry has been emerging among the manufacturers of industrial engines in recent years. In particular, small captive manufacturers – whose annual engine production is 20,000–30,000 units – are increasingly switching to purchasing their engines on the non-captive market.

18.2.3 Applications

The market for stationary and industrial engines is characterized by a multitude of applications that are very frequently connected with low quantities down to job production. The art of engine manufacturing consists of covering this multiplicity of applications without sinking into an unmanageable and commercially unsustainable diversity of variants in one's own production in the process. The solution is a platform strategy based on a basic engine and a modular concept for the add-on components as pictured in Fig. 18-12 [18-1]. Installation conditions have given rise to a particular diversity of variants for oil pans, intake manifolds and exhaust manifolds. The latter must accommodate the various mounting positions for exhaust gas turbochargers. Alternators may be mounted just as variably when their installation is required in the first place.

The multiplicity of applications not only ensues from the installation restrictions but also the load profiles, climatic operating conditions, fuel grades, different emission standards, fuel consumption requirements and the sales price obtainable for an engine.

While only engines that meet the requirements of emission level 3 (see Sect. 15.2) are still sold in the European Union and North America, large parts of Africa and the Middle East have no emission standards of any sort in force. The sale of engines compliant with emission level 3 is impossible in these regions for commercial reasons and because of the engines' technological complexity. Air or oil-cooled engines with mechanically controlled injection systems are also preferred here for climatic and logistical reasons.

The widely varying grades of fuel throughout the world are another aspect that already has to be taken into account during development. While a fuel standard compliant with EN590 with a cetane number of at least 51 applies in Western Europe, diesel fuels in the USA have an average cetane number of 40–42. Experience has shown that this increases nitrogen oxide emissions by approximately 0.2 g/kWh. Since the same limits for industrial engines apply in the USA and Europe, this must be factored in when an engine is modified. Furthermore, the fuel's sulfur content – above 5,000 ppm in some regions – represents a challenge both in terms of the effects on particulate emissions and damage by sulfuric acid corrosion. Some EU countries allow the use of fuel oil for industrial engines. This is not compliant with the EN590 standard either. Biofuels are increasingly being used. Subsumed under the term FAME (fatty acid methyl ester), such esterified vegetable oil-based fuels are normally not approved by manufacturers of injection systems. Thus, engine manufacturers bear the risk of approval. Rape oil methyl ester (RME) is the biodiesel common in Germany (see Sect. 4.2).

Since an engine's useful life also depends on its operational demands, manufacturers calibrate them in performance classes (see Table 18-2). An engine's power is reduced to prevent thermal overloading by steady full load operation.

The cooler package for a tractor illustrates the complexity of installation in Fig. 18-13. The installation accommodates the tractor's down swept front end, the design of which improves the driver's field of view. Since cooler efficiency quite heavily depends on configuration and flow, installation must be simulated before technical approval to ensure that the cooler package conforms to the engine's design data. This is necessary to comply with emission requirements as well as to prevent the engine from thermally overloading.

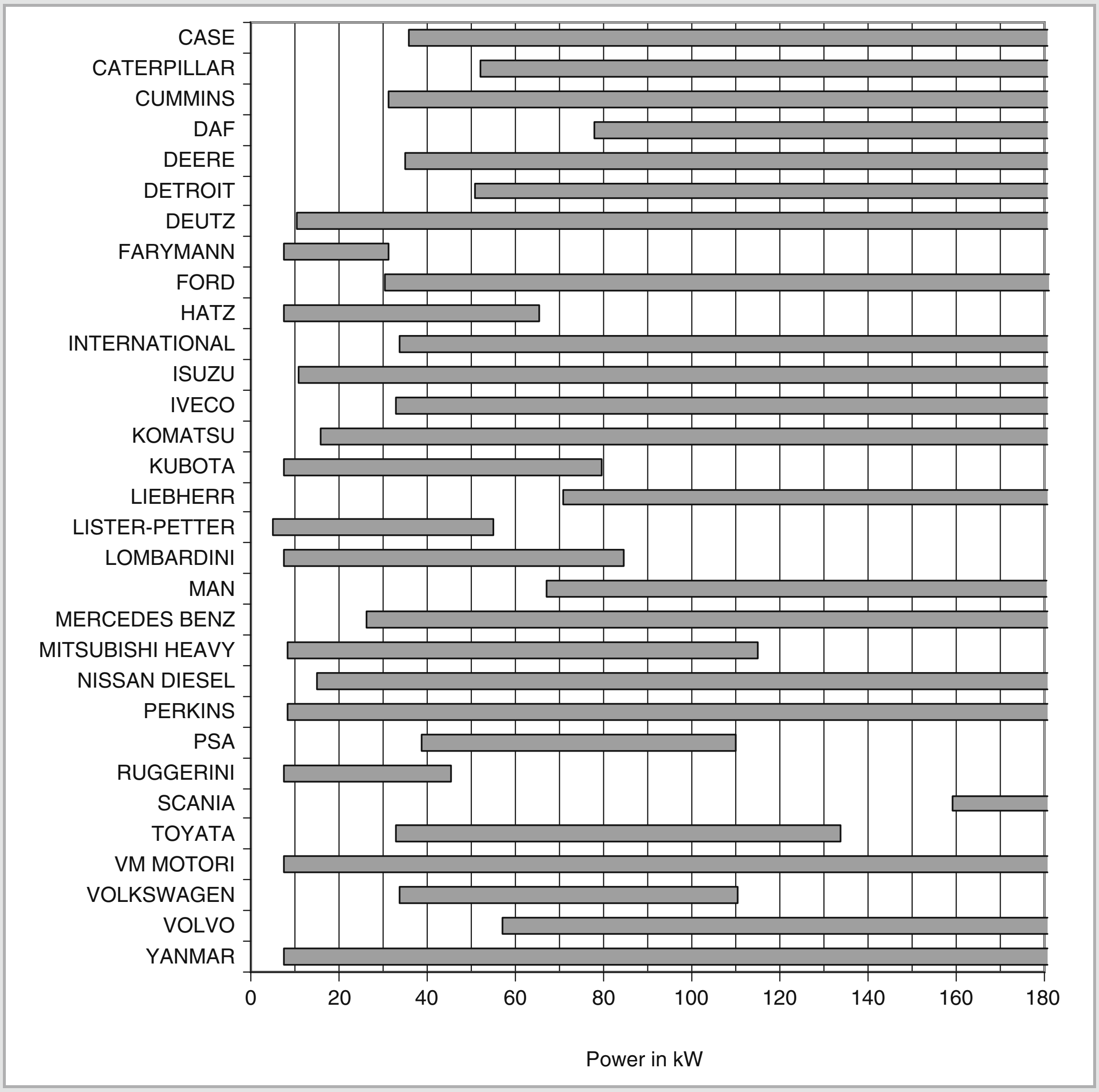

Fig. 18-11 The range of industrial engine power from the most important (German and foreign) manufacturers (source: DEUTZ AG market analysis)

18.2.4 Modified Vehicle Engines

18.2.4.1 General Remarks

Along with the special diesel engines developed solely as industrial engines, other modified car or commercial vehicle engines also end up in general industrial use. The indisputable advantages of modified vehicle engines are the cost advantage generated by synergies with mass production and lightweight construction with a good weight-to-power ratio.

Limits on their application arise whenever cost effectiveness or technical features suffer, e.g. overloading of the crankshaft assembly, which is distinguished by lightweight construction in vehicle engines. Therefore, the suspension of a weight-optimized engine ought to be treated just as if it were installed in a vehicle. For reasons of strength, an engine crankcase should not to be drawn on for system-supporting functions as is frequently common in the design of agricultural and construction equipment. Since a weight-optimized vehicle engine has less mass, its noise and vibration damping measures already require somewhat more complex treatment

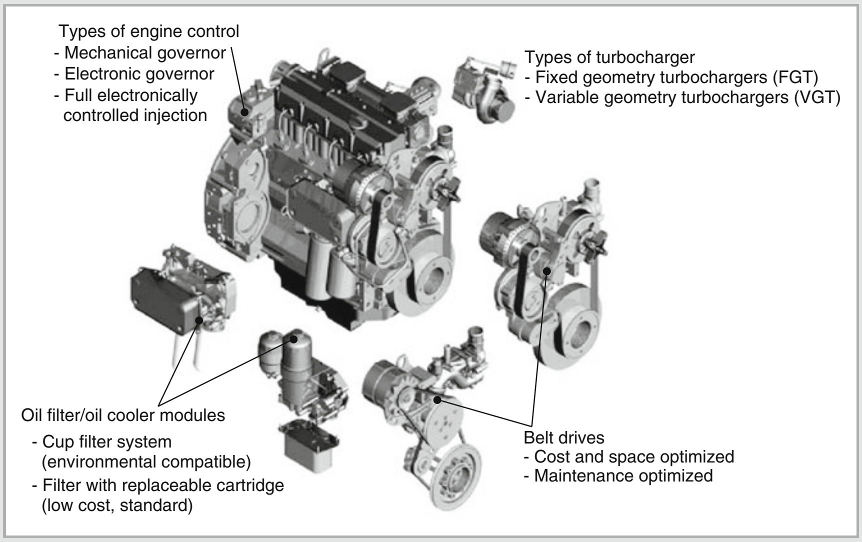

Fig. 18-12 Modular concept of an industrial engine consisting of the basic engine and various add-on part options

Table 18-2 Definition of performance classes and examples of respective applications

Power class	Power reduction %	Vehicle engines	Stationary engines		
			Construction equipment	Agricultural and forestry equipment	Pumps and compressors
I	0	Construction site vehicles Fire vehicles Dump trucks Crane vehicles Street cleaners	Front loaders Backhoe loaders Graders Earth moving equipment Road rollers Concrete and mortar mixers	Combine harvesters	Fire pumps Emergency pumps
II	5	Snow blowers Snowplows	Hydraulic excavators Blacktop paver Concrete and road milling machines	Four-wheel tractors Skidders Pruning platforms Chopper forage harvesters Harvesters	Sprinkling and irrigation systems High pressure compressors up to 10 bar
III	10		Trench diggers Drilling equipment		High pressure compressors over 10 bar

than a heavy industrial engine's. A vehicle engine's power is adjusted to requirements specific to the vehicle and, in accordance with DIN/ISO 3046, should be selected significantly lower for industrial engines in continuous operation to facilitate engine service life with low wear (see Sect. 18.2.3). When high starting torque is required, a better commercial vehicle engine rather than a car diesel engine ought to be the stated basis for engine selection from the start: Car engines normally do not reach their maximum torque until the upper speed range. While special adjustment of injection pumps enables shifting the maximum torque to the lower speed range within certain limits, any deviation from standard vehicle equipment becomes expensive and unprofitable when it requires too much special equipment, which may well be desirable but is not absolutely necessary.

In order to be able to assess them, the options for modifying a standard vehicle engine for needs when used as an industrial engine have to be harmonized with the requirements.

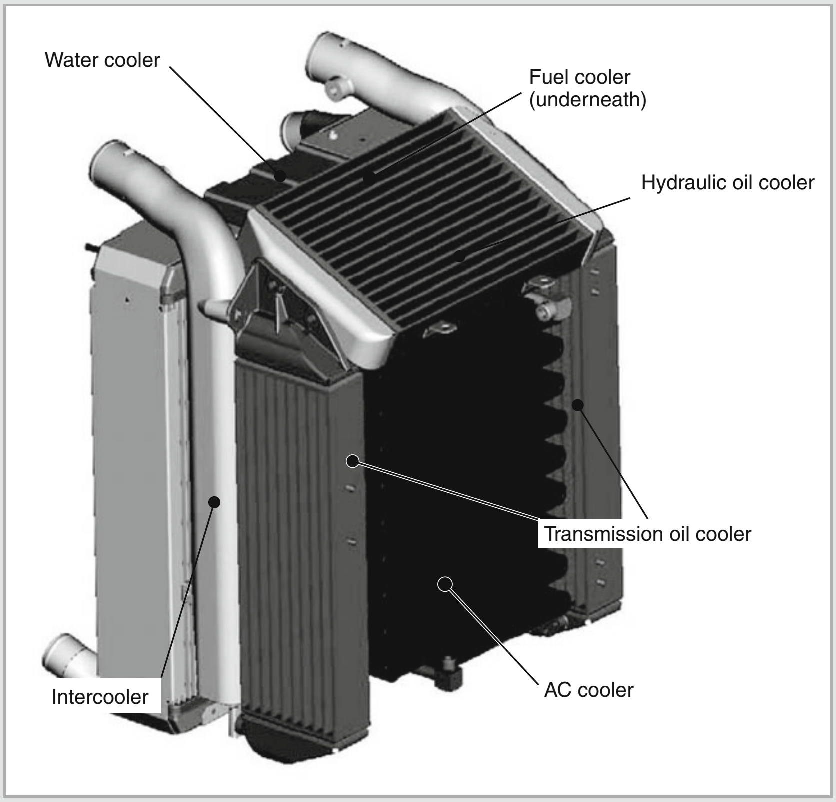

Fig. 18-13
Tractor cooler package with seven cooling modules

Service life and maintenance intervals play a crucial role for industrial engines. Such installation often requires larger engine oil volumes to facilitate longer oil maintenance intervals. Matched to the conditions of installation, special oil pans with an oil volume of up to 20 l are employed. The car version is approximately 4 l. Not only the extra costs, but also existing manufacturing facilities and manufacturing options have to be weighed to determine whether such a modification still leads to the desired aims or is even affordable.

18.2.4.2 Implemented Engines

Figure 18-14 pictures a modified vehicle engine manufactured by Volkswagen [18-2]. The engine is based on a car engine, the only difference being is its dataset to obtain maximum synergy with mass car production. Swirl chamber engines are still manufactured as naturally aspirated engines only in the power range up to 37 kW. However, stage 3A emission legislation requires that they be replaced by direct injection engines, which are manufactured both as naturally aspirated engines and as turbocharged engines with and without intercooling up to approximately 80 kW with displacements of 1.9 and 2.5 l.

A dataset concept consisting of a block of the particular basic engine functions independent of the application and a block of industrial engine functions was developed to modify these basic engines developed for vehicle use for the widest variety of requirements for industrial applications. The engine control unit stores the industrial engine functions in seven specific datasets. These differ in their type of engine control:

– torque control specified by the accelerator pedal,
– power control specified by the accelerator pedal,
– operating speed control specified by the accelerator pedal taking the form of a proportional governor,
– operating speed control by a 0–5 V interface taking the form of a proportional integral governor with or without a safety concept,
– automotive driving by converting driver demand by the accelerator pedal into an injected fuel quantity and
– stationary operation by an externally connected fixed speed governor.

Diverging from vehicle engines, the electronics has a few actuators and sensors specifically for industrial applications. The variance described here has only been producible in engines with a mechanical-hydraulic injection pump by

- Direct injection
- Distributor injection pump VP 37
- ECU Bosch EDC 15V
- SOHC valve train
- Aluminum cylinder head
- Vertical rechargeable cartridge oil filter
- VGT
- TDI 2.5 specifications:
  - 80kW at 3.500 rpm
  - 280Nm at 1,400 - 2,400 rpm
  - 200kg
  - 211 g/kWh(bsfc $_{best}$)

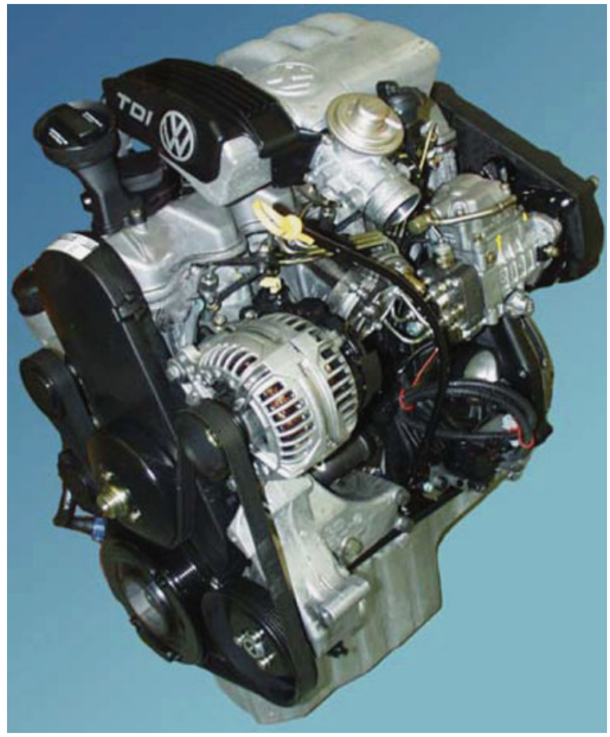

Fig. 18-14 VW supercharged 2.5 l TDI® engine with electronically controlled distributor injection pump, two-valve cylinder head, variable turbine geometry turbocharger and a maximum power of 80 kW at 3,500 rpm

setting the injection pump as a function of the application in conjunction with special equipment such as a variable-speed governor and frequently only together with an additional electronic governor.

18.2.5 Industrial Engines

18.2.5.1 Product Concept

Diesel engines exclusively developed as stationary and industrial engines can be found in the performance class up to approximately 75 kW. All numbers of cylinders between one and four are common. Fuel consumption and power density play a subordinate role; procurement costs, ruggedness and versatility are more important. Naturally aspirated engines are increasingly being replaced by supercharged engines because of emission requirements but frequently without an intercooler for reasons of installation and cost. However, naturally aspirated engines still dominate the power range up to 37 kW. Several power take-offs and the option of 100% power take-off on the damper side crankshaft end are common. The low maintenance requirements have led to a very large proportion of engines in this performance class being air or oil/air-cooled.

Also implementable as vehicle engines, industrial engines in the power range above 75 kW – designed as four, six or eight cylinder engines – are usually either derived from commercial vehicle engines or industrial engines. The requirements for exhaust and noise emission, maintenance, service life, weight-to-power ratio, power density, etc. for industrial engines in this performance class are similar to those for commercial vehicle engines. The only difference from commercial vehicle engines are special design features such as power take-offs including take-offs on the damper side crankshaft end too, particularly rigid engine blocks, stiffened oil pans and engine balancers when installed in tractors, additional cooling systems for hydraulic systems and generators.

The variety of potential uses for industrial engines and thus potential customer demands necessitate just as many equipment options, which often may only be produced in relatively small quantities. Hence, industrial engines are normally more expensive than corresponding car or commercial vehicle engines.

18.2.5.2 Implemented Engines

Given the broad range of industrial engines, this section presents the basic features of a few examples implemented in the segment for displacements between 200 and 400 cm^3 and the segment in the range of one liter per cylinder. Space limitations preclude coverage of the vast variety.

Virtually only designed as inline engines, a multitude of multiple cylinder water-cooled engines occupy the segment for small displacements up to 400 cm^3. Important manufacturers are Kubota, Yanmar, Daihatsu, Isuzu, Lister Petter and Deutz. Their basic design principles are all very similar. However, the injection equipment varies widely from pump-line-nozzle systems to distributor pumps and inline pumps up through unit pump systems. The direct injection swirl chamber principle is frequently employed. Along with its noise advantage and lower manufacturing costs, it also has a lower level of nitrogen oxide emission than the direct injection principle. The disadvantageous fuel consumption connected with it is accepted in this market segment.

A very interesting example of design is Hatz's W35 series (see Fig. 18-15) with a displacement of 350 cm^3 per cylinder and available as an inline engine with two, three and four cylinders. This inline engine has a vertical parting plane running longitudinally. Both halves of the pressure die cast aluminum crankcase already contain an oil pan, timing case, gear case, flywheel housing, cylinder head and cylinder liner holder. The cylinder head and the thin-walled centrifugally cast cylinder liner are modules. The water pump is integrated in the timing case's gear drive. The engine is already completely enclosed with merely three components, i.e. the left and right crankcase half and the valve cover. The injection system and its timing are modularly designed. A mechanically controlled, very compact unit injector is driven by the overhead camshaft and the driving lever. Figure 18-16 pictures the design of a completely equipped 4W35 naturally aspirated engine.

The implemented examples in Figs. 18-17 and 18-18 illustrate the ways different market and statutory requirements influence the design of an engine series. An emission limit of 4.7 g/kWh for NO_X+HC and 0.3 g/kWh for particulate matter in the C1 cycle compliant with ISO 8178 applies to the 64 kW version (see Sect. 15.2). For reasons of space and cost, this engine does not have an intercooler and is equipped with a mechanically controlled unit pump injection system. The emissions goal was achieved by appropriately designing the combustion chamber and injection nozzle geometry, injection timing and gas exchange. The disadvantages of fuel consumption connected with this concept are more than compensated by the low manufacturing costs. The 113 kW version in Fig. 18-18 was modified for high capacity utilization in a tractor. An emission limit of 4.0 g/kWh for NO_X+HC and 0.2 g/kWh for particulate matter in the C1 cycle applies to this power. Fuel consumption plays a significantly larger role because of the load profile and the high number of operating hours per year. More technological complexity and thus a higher engine price are accepted. Both intercooling and cooled exhaust gas recirculation are suitable measures to obtain low nitrogen oxide emissions in conjunction with good fuel consumption. In addition, the common rail injection system allows optimizing fuel consumption, combustion noise and emissions in the entire engine map [18-3, 18-4].

These typical stationary engines furnish more options for power take-off. The timing gear drive can drive not only the camshaft but also one or more hydraulic pumps and/or compressors. Alternatively, up to 100% of the engine power can also be taken off the front side. These water-cooled engines are also provided with integrated cooling systems, the oil and water cooler being attached on the sides of the engine. An additional cooler for hydraulic oil can also be provided as desired. This not only facilitates extremely compact design but also simplifies the assembly of a complete engine delivered with integrated cooling.

Agricultural machinery is a typical case of industrial engine application. The high rigidity of the engine block not only diminishes noise emission but also enables utilizing the engine as a supporting tractor component. In the four cylinder version, a balancing differential gear eliminates the free inertial forces and thus guarantees the smooth operation necessary for rigid installation in a tractor.

18.2.6 Outlook

Like the development of vehicle engines, the further development of industrial engines will be characterized in coming years by a further drastic tightening of emission control legislation (see Sect. 15.2). The related technological requirements in stage 3A emission legislation have already led to the replacement of mechanically controlled injection systems by electronically controlled systems in many engines. Mechanically controlled injection systems only continue to be dominant in the power range below 75 kW.

Compliance with limit level 4 requires the introduction of exhaust aftertreatment technologies such as particulate filters and nitrogen oxide aftertreatment by SCR. Since these technologies require monitoring and control, integrated engine management that also takes over the control of the exhaust temperature for exhaust aftertreatment is indispensible. In view of the higher emission limits, low cost mechanically controlled injection systems will probably only be applied in the power range below 56 kW in the future too.

The common rail injection system is considered to have the greatest future potential because of its flexibility, particularly the option of multiple injections. This also simplifies the modification of dynamic operating performance, which

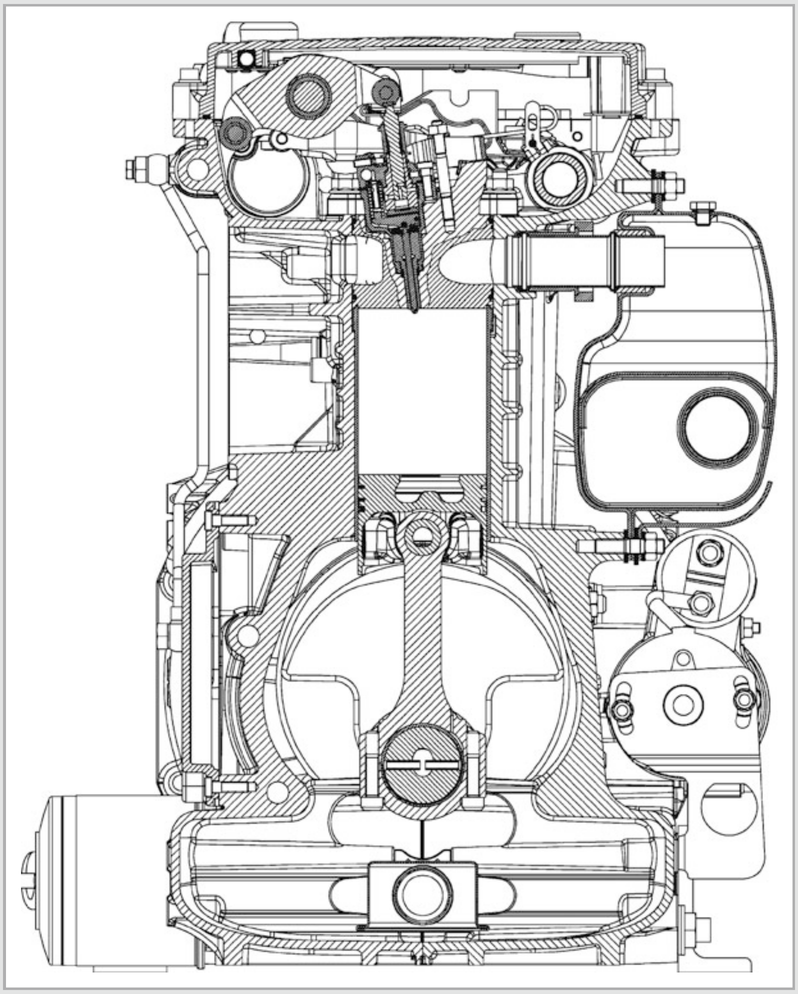

Fig. 18-15
Cross section of Hatz's 4W35NA engine. Vertical longitudinally split pressure die cast aluminum crankcase

must not only satisfy customer demands but also emission control legislation by introducing a transient cycle. In addition, electronic engine management furnishes the option of integrating additional customer functions and allows data exchange with other vehicle or machine systems.

This increasing complexity of technologies and their integration in the overall engine system not only demands engine manufacturers have expertise in the field of injection and combustion but also additional expertise in electronics and exhaust aftertreatment. Machinery manufacturers that produce engines in relatively small quantities for internal need are hardly able to do this. They are increasingly switching to buying engines from manufacturers that, by virtue of their size, have core competencies in the aforementioned fields. Thus, a continuation of the concentration process among engine manufacturers may also be expected in the coming years.

18.3 Medium Speed Four-Stroke Diesel Engines

18.3.1 Definition and Description

18.3.1.1 Classification of Medium Speed Four-Stroke Diesel Engines

The term medium speed engines refers to trunk-piston engines that now almost exclusively operate with the four-stroke process. Still sporadically available on the market, two-stroke trunk-piston engines are of secondary importance and are not examined here any more closely.

Medium speed four-stroke engines have speeds of between approximately 300 and 1,200 rpm. Their cylinder dimensions range from under 200 to over 600 mm. In recent years, mean piston velocities have settled at approximately 9–11 m/s and even more in individual cases.

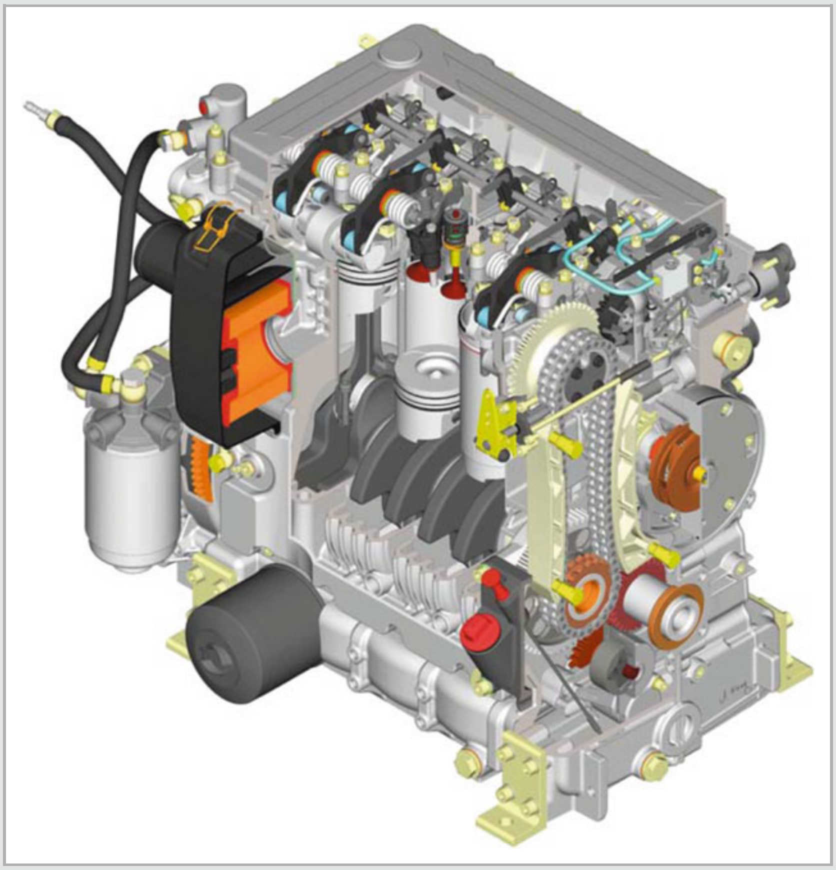

Fig. 18-16
Hatz W35 series water-cooled four cylinder OHC inline engine with mechanically controlled unit pump

In addition, brake mean effective pressures p_e of up to 29 bar are being attained. This corresponds to effective brake work of $w_e \leq 2.9$ kJ/dm^3.

These values yield a power range for modern medium speed four-stroke engines of approximately 100 to over 2,000 kW/cylinder.

Inline engines are now built with six to ten cylinders (sometimes even fewer) and V engines with twelve to twenty cylinders.

18.3.1.2 Medium Speed Diesel Engine Use

Medium speed four-stroke engines are employed as main marine engines, auxiliary marine engines and stationary engines to drive generators. Smaller medium speed engines are also implemented to drive pumps and compressors, in combined heat and power stations or as traction motors.

In the last forty years, diesel engines have established themselves in civil shipping over other propulsion alternatives (steam or gas turbines). Medium speed four-stroke engines have gained great importance and become established for types of ships that were previously reserved for two-stroke low speed engines (Fig. 18-19). Medium speed engines are the only option for many cases of application from the outset for reasons of space utilization, e.g. ferries, RoRo ships and other special ships. Diesel-electric drives with medium speed engines are frequently implemented in passenger and cruise ships because of their high flexibility. Smaller medium speed engines are predominantly employed as driving engines in inland shipping too. At least in larger ships, medium speed engines are used almost exclusively as auxiliary marine engines (Fig. 18-20).

Another field of application is diesel power stations that generate electricity, which are extremely widespread, particularly in industrial threshold countries without interconnected power systems covering entire regions. An appropriate number of machine sets with unit outputs of 10–20 MW can be used to cost effectively erect power stations of up to 100 MW and higher in short time, especially since they can be progressively expanded based on demand.

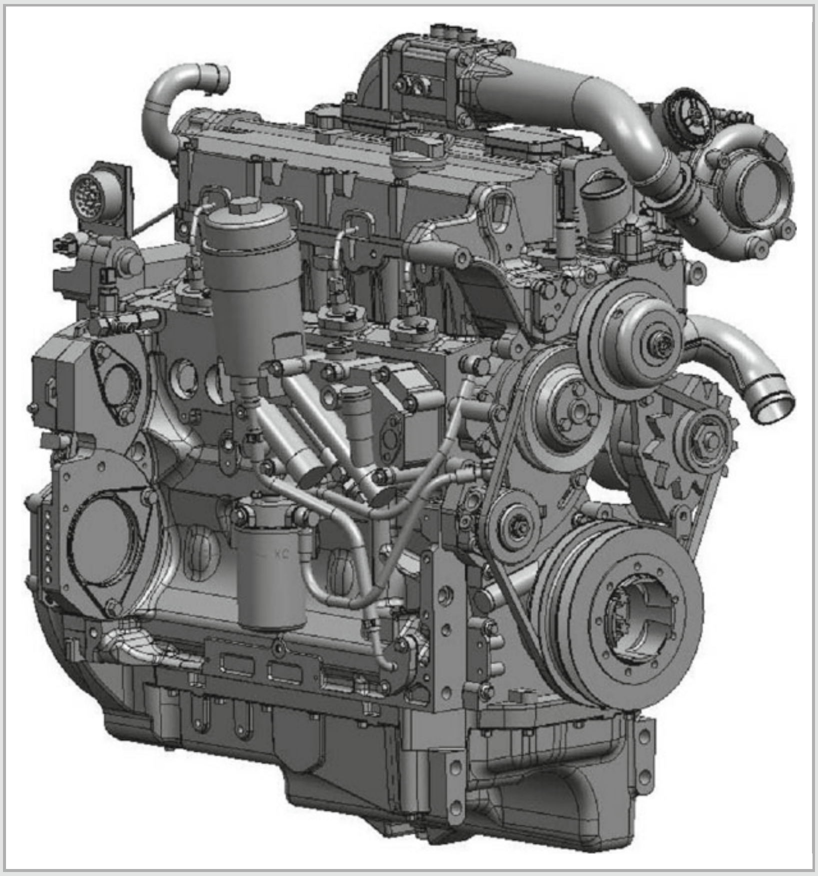

Fig. 18-17
DEUTZ TD2012L04 2 V turbocharged four cylinder engine without intercooler and with mechanically controlled pump-line-nozzle injection system, two-valve cylinder head, side-mounted exhaust gas turbocharger. Maximum power of 67 kW at 2,200 rpm

18.3.1.3 Fuels

Heavy Fuel Oil Operation

Advanced larger medium speed engines are able to process even poor grades of heavy fuel oils as defined in CIMAC H/K55 (see Sect. 4.3). This has been achieved by consistently developing components, above all the combustion chamber, valves, pistons and cylinder liners, etc. Heavy fuel oil is used almost exclusively in large medium speed engines that are utilized both as main marine engines and in stationary plants, provided environmental requirements do not impose any restrictions.

The time available for combustion is a significant variable that influences the combustion of heavy fuel oils. Understandably, larger medium speed engines with speeds of up to 750 rpm are more easily built to be suitable for heavy fuel oil than smaller engines with speeds in the range of 1,000 rpm and above. The permissible grade of heavy fuel oil may generate certain restrictions for such higher speed engines. Nevertheless, auxiliary marine engines are also increasingly being designed for heavy fuel oil operation to be able to supply main and auxiliary machines with the same fuel (unifueled ships).

Appropriate precautions must be taken for the combustion of heavy fuel oil, both to process the fuel and to select the lubricating oil (see Sect. 4.3). Engine design must ensure that the components influenced by the heavy fuel oil have the "right" temperatures to prevent harmful effects of corrosion and deposits (see also Sects. 18.3.3 and 7.1).

Hence, smaller higher medium speed engines are often run with diesel oil. The time and effort required to process heavy fuel oil is not worthwhile in most cases.

Gas Operation

Along with heavy fuel oils and the widest variety of grades of diesel fuels, medium speed four-stroke engines are also run with different fuel gases. Both spark ignited and dual fuel systems are employed (see Sect. 4.4).

Unlike diesel operation, conventional dual fuel design in which a homogeneous air/fuel mixture is compressed and ignition is triggered by injecting a small quantity of ignition

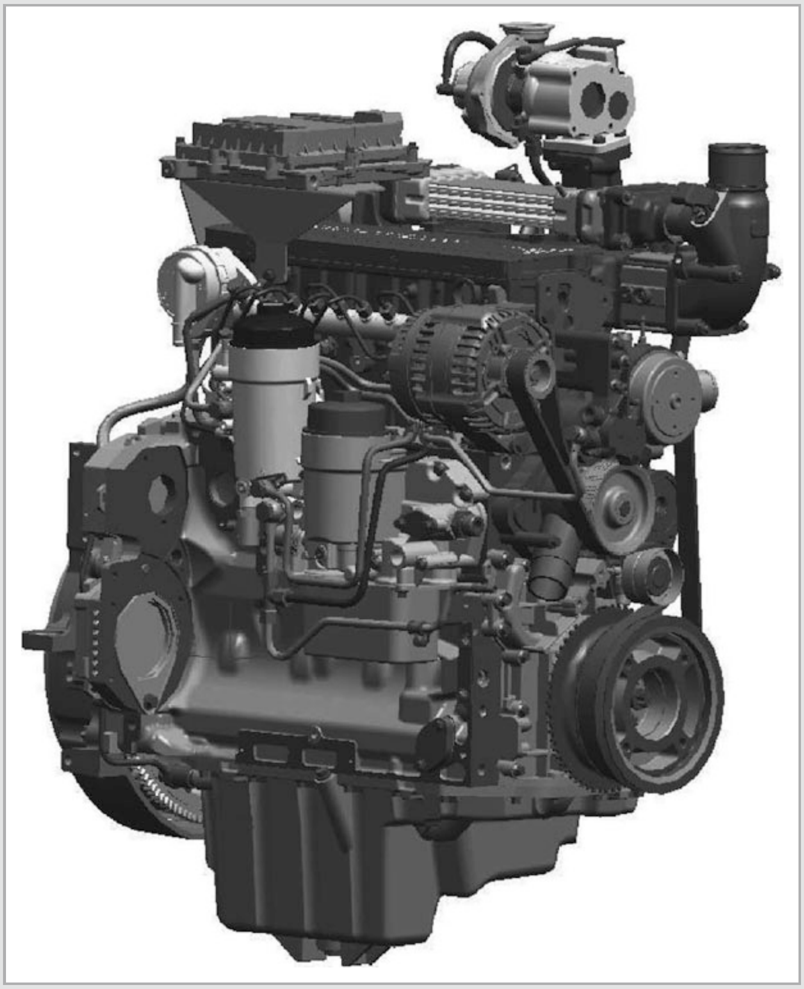

Fig. 18-18
DEUTZ TD2012L04 2 V supercharged four cylinder engine with intercooling and electronically controlled common rail injection system, four-valve cylinder head, centered overhead exhaust gas turbocharger and external cooled exhaust gas recirculation. Maximum power of 113 kW at 2,200 rpm

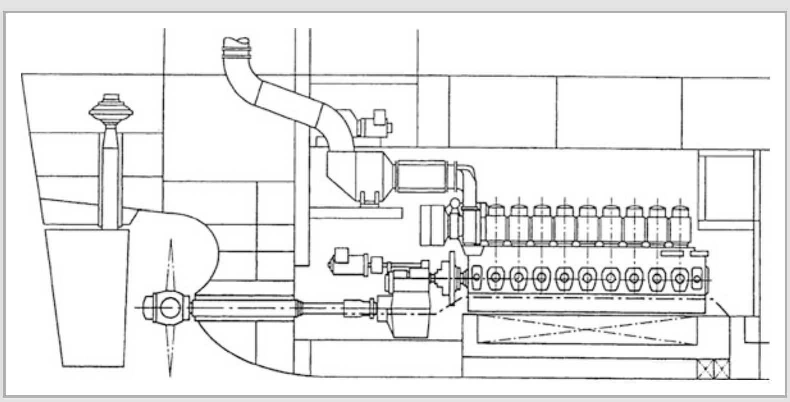

Fig. 18-19
Single engine system with a medium speed engine, reduction gear and generator driven by PTO (power take-off). Engine power of 12,500 kW at 428 rpm to drive a refrigerated cargo ship with a rated capacity of 400,000 cft

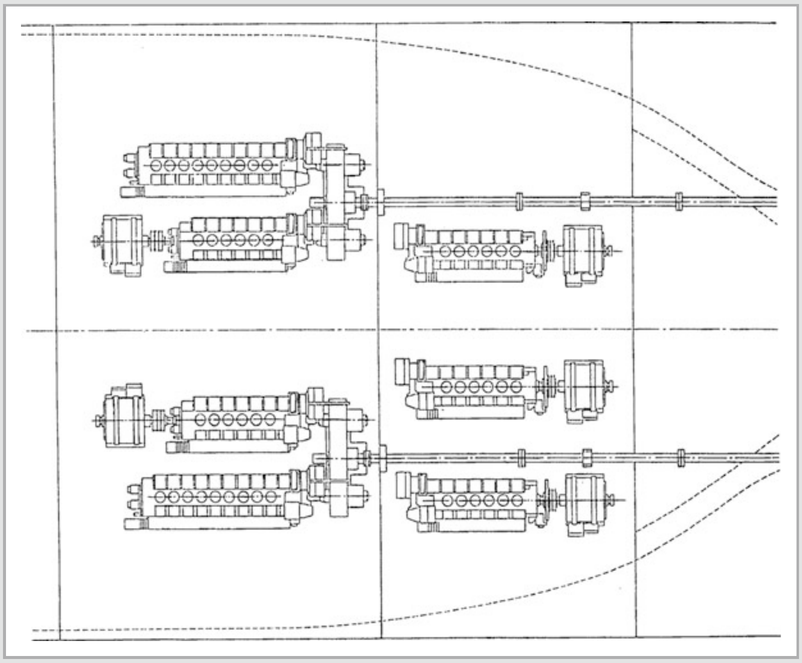

Fig. 18-20
Cruise ship machine room configuration: each of the main engines is in a double "father-and-son" configuration using a reduction gear to operate two propellers. There are three auxiliary engines of the same type to generate on-board power and additionally generators driven by the "son" engines

oil must recover the brake mean effective pressure and the power. As recent developments have demonstrated [18-5], appropriately designed dual fuel engines have come close to the power of diesel operation without having to apply the dual fuel process with high pressure gas injection (see Sect. 4.4.3.1).

The lean burn process was developed to keep emission values low. Ignition oil ignites a lean air/fuel mixture in a secondary combustion chamber. Upon being discharged from the prechamber, it serves as a high energy ignition aid for the lean mixture in the main combustion chamber [18-6].

Both the lean burn system and spark ignited engines have become increasingly widespread among medium speed four-stroke engines in recent years.

18.3.1.4 Advantages of Medium Speed Engines

Medium speed four-stroke diesel engines are situated between high speed high performance engines and two-stroke low speed engines. In terms of use, the transitions are fluid.

The basic advantages of medium speed engines over two-stroke engines are their lower space requirement and comparatively low weight-to-power ratio in conjunction with better specific costs [18-7].

This holds true even though a medium speed propulsion engine is always equipped with a reduction gear (Fig. 18-21).

Apart from the space advantages, there are other points in favor of medium speed engines:

- free selection of the optimal propeller speed,
- good suitability for elastic installation to insulate structure-borne noise,
- very simple option to recover waste heat,
- speeds common to generators,
- simple shaft generator attachment to generate power in heavy fuel oil operation,
- good prerequisites for measures that reduce pollutants,
- easy attachment of power and compound turbines to increase cost effectiveness and
- good suitability for engine management systems and remote monitoring.

18.3.2 Design Criteria

18.3.2.1 Specific Power

Triggered by competitive pressure and facilitated by the further development of supercharging equipment, specific powers have continuously been increased over the course of time. Brake mean effective pressures and specific work have partly reached the limit of what is achievable with one-stage supercharging. Mean piston velocity has also been increased continuously.

Proportional to the product of specific work and mean piston velocity, the specific power per unit piston area P_A, (see Sect. 1.2) is a parameter for the state-of-the-art. Today's medium speed four-stroke diesel engines attain

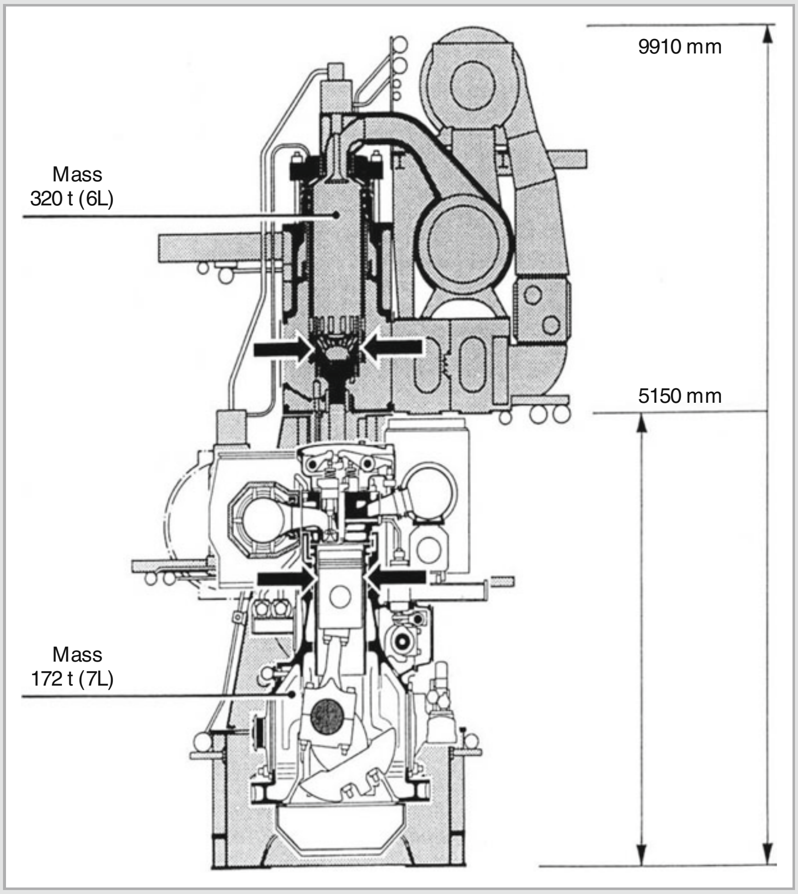

Fig. 18-21
Size comparison of a medium speed four-stroke engine and a two-stroke low speed crosshead engine of equal power

powers per unit piston area of 5 W/mm^2 at peak values of approximately 7 W/mm^2.

18.3.2.2 Maximum Cylinder Pressure

Parallel to increasing specific power per unit piston area, great efforts have been made to lower fuel consumption, not least motivated by the oil crises in the 1970s and 1980s and the steady rise of fuel prices. The ratio of maximum cylinder pressure to effective brake work, also expressed by the ratio p_{Zmax}/p_e, has proven to be an important value to characterize efficiency. When possible, peak pressure also had to be elevated disproportionately as mean pressure increased to assure a sufficiently high p_{Zmax}/p_e ratio of approximately 7–8 and, thus, minimum fuel consumption. Hence, maximum cylinder pressures have reached a remarkable level in recent years. Engines with peak pressure of 200 bar are already in operation and the trend is continuing. Figure 18-22 presents the development of maximum cylinder pressure and specific fuel consumption over the past decades.

18.3.2.3 Stroke to Bore Ratio

It is important that the rate of pressure increase $dp_Z/d\varphi$, i.e. the interval between final compression pressure and maximum cylinder pressure or so-called ignition jump, does not grow too large, particularly in heavy fuel oil operation. It follows from this that high maximum pressures also require significantly higher final compression pressures. The final compression pressure is influenced by the boost pressure and compression ratio. The boost pressure level is limited for reasons of thermodynamics (see Sect. 2.2). The permissible boost pressure can be expediently described with a parameter that specifies the ratio of boost pressure to brake mean effective pressure. A p_L/p_e ratio of 0.15–0.17 has proven optimal for medium speed engines to ensure consumption is low on the one hand and to keep the temperature level of the combustion chamber components in heavy fuel oil operation in a safe operating range on the other. Finally, it follows that the compression ratio must be elevated commensurately to obtain the desired final compression pressure. Thus, as a

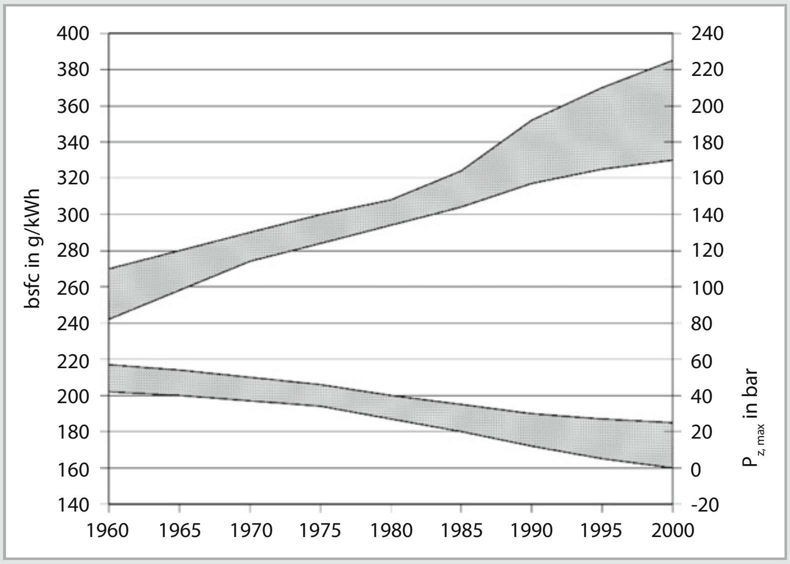

Fig. 18-22
Development of maximum cylinder pressure and specific fuel consumption in medium speed four-stroke engines over the last 40 years

function of the bore diameter, the compression ratio of present day medium speed engines is $\varepsilon = 13–16$.

A higher compression ratio can be obtained more easily with a longer stroke engine as well as with a well shaped combustion chamber. As the stroke grows shorter, the combustion chamber becomes flatter and flatter at the specified compression ratio and good combustion becomes increasingly difficult to obtain.

Absolute engine size also plays a role in all these considerations. The smaller the cylinder dimensions become, the more adversely detrimental spaces around the valves make themselves noticeable. They increase disproportionately as dimensions grow smaller. Logically, a smaller stroke to bore ratio suffices to obtain a specific compression ratio when the cylinder diameter is larger but not when it is smaller.

18.3.2.4 Speed

Piston stroke and mean piston velocity produce the appropriate engine speed. Speeds in the range of approximately 300–1,200 rpm can be produced depending on the cylinder diameter, stroke to bore ratio and maximum permissible mean piston velocity. Thus, appropriate generator speeds can power engines that generate three-phase current with 50 Hz or 60 Hz (see Sect.1.2).

18.3.2.5 Other Criteria

In addition to medium speed four-stroke engines' low fuel and lubricating oil consumption, suitability for heavy fuel oil, good manufacturing costs, etc., operators also place great value on simplicity of assembly and ease of maintenance.

This is one reason complex technical solutions, e.g. one or multiple stage supercharging, have not become established among medium speed engines.

Emphases of development are suitability for heavy fuel oil even under high specific loads as well as cost effectiveness, reliability and improved exhaust emission. Examples of design solutions illustrate this in the following section.

18.3.3 Design Solutions

18.3.3.1 Basic Engine Design

Space limitation only permit touching on a few basic components and describing their principle features here.

The formerly frequently common crankcase design with a bedplate, a crankshaft inserted from above and an externally mounted cylinder block bolted to the bedplate with tensioning bolts has been replaced in most cases with newer designs with a one-piece frame design with an overhead crankshaft. This design assures a very good load transfer, eliminates additional loaded interfaces and is inexpensive.

MAN Diesel chose an interesting solution for medium speed engines. Elongated main bearing bolts running to the top edge of the one-piece frame and cylinder cover bolts extending deep into the frame significantly relieve the load on the cast structure (Fig. 18-23).

18.3.3.2 Crankshaft Assembly

Crankshaft and crankshaft bearing. In addition to appropriate oil care, the sizing of the crankshaft bearing is extremely important for the prevention of bearing problems in

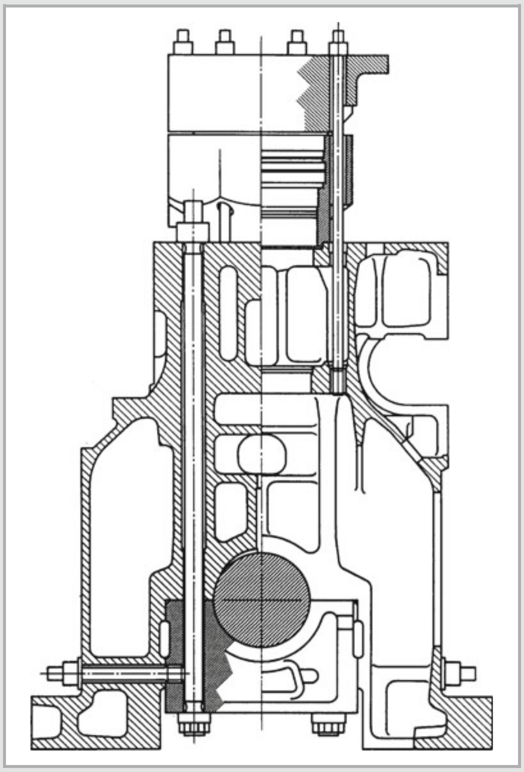

Fig. 18-23 Engine frame with single cylinder jackets, elongated main bearing bolts and elongated cylinder cover bolts (source: MAN Diesel)

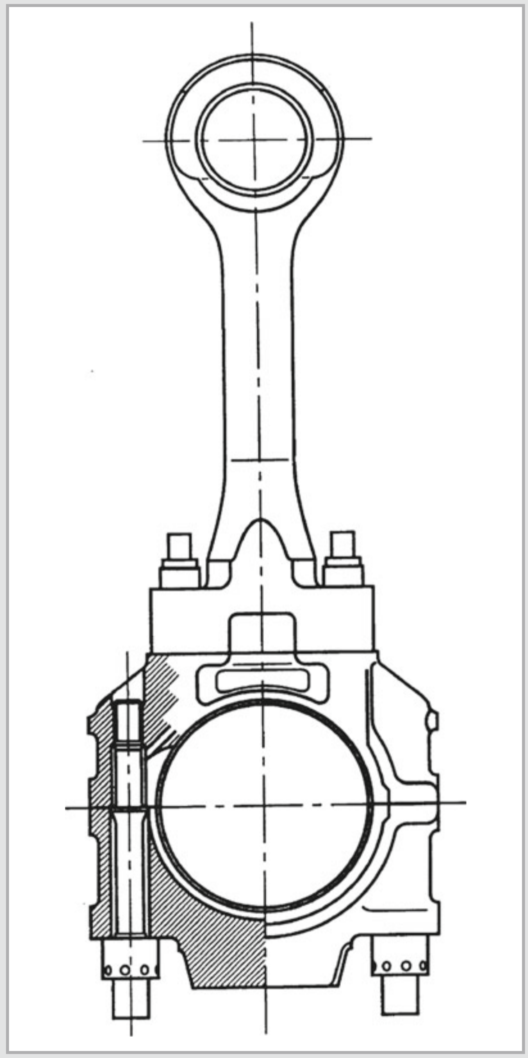

Fig. 18-24 Connecting rod with marine head design

heavy fuel oil operation, which are caused by corrosive or abrasive wear. Practice has demonstrated that a certain minimum residual gap in the lubricating film may not be undershot if satisfactory service lives of the bearing shells are to be obtained. Significantly higher than earlier, the cylinder pressures frequently necessitate stronger basic bearings and crankshaft journals that enlarge the bearing area for reasons of strength. Moreover, the introduction of new bearing technologies, e.g. grooved bearings or sputter bearings, has increased stability considerably. Despite the higher gas forces, this has even enabled significantly increasing the operational reliability of the suspension and the service lives of the bearings in many cases (see Sect. 8.5).

C**onnecting rod.** The crankshaft journal bearing may be split simply in a straight line only in the fewest cases when engines have comparatively low loads. As a rule, stronger crankshaft journals matched to higher loads require that the connecting rod be split obliquely in order to at least be able to guide the connecting rod and the connecting rod shank through the cylinder liner when the piston is pulled. A marine head design in which the shank is bolted with its own two-piece bearing body is employed in many cases (Fig. 18-24). This additional parting line has the advantage of fewer spatial limitations when sizing the bearing body and, hence, facilitates a rigid low-deformation design. Moreover, the bearing does not have to be opened when a piston has to be disassembled.

18.3.3.3 Combustion Chamber Components

Piston. In addition to monoblock nodular cast iron pistons for smaller cylinder dimensions, medium speed four-stroke engines have composite pistons in most cases. The piston top part is steel and the ring grooves are often hardened or chrome plated to reduce wear (see Sect. 8.6).

The piston skirt predominantly consists of nodular cast iron because of the increased load or less frequently of a light alloy. The piston skirt and piston crown are sporadically made of steel [18-8]. Such composite steel/nodular cast iron pistons make it possible to control firing pressures of up to over 200 bar.

The high thermal load requires optimal cooling of the piston top part (see Sect. 7.1). Oil from the circulation system, which is usually supplied to the piston through the connecting rod, is employed as the cooling medium (see Sect. 8.6). The shaker effect during the piston's upward and downward motion flings the cooling oil onto the inner walls of the piston crown where it absorbs the heat and then returns to the driving chamber through appropriate return bores in the piston skirt. The piston top part is often furnished with cooling bores to enlarge the area of heat transfer (Fig. 18-25).

In conjunction with a piston top land ring on the cylinder liner, the piston is designed as a stepped piston to prevent deposits of combustion residues and thus bare spots on the cylinder liner and to additionally reduce oil consumption. Narrow piston clearance that traps abrasive particulates and protects the lubricating film reduces the piston rings' mechanical load.

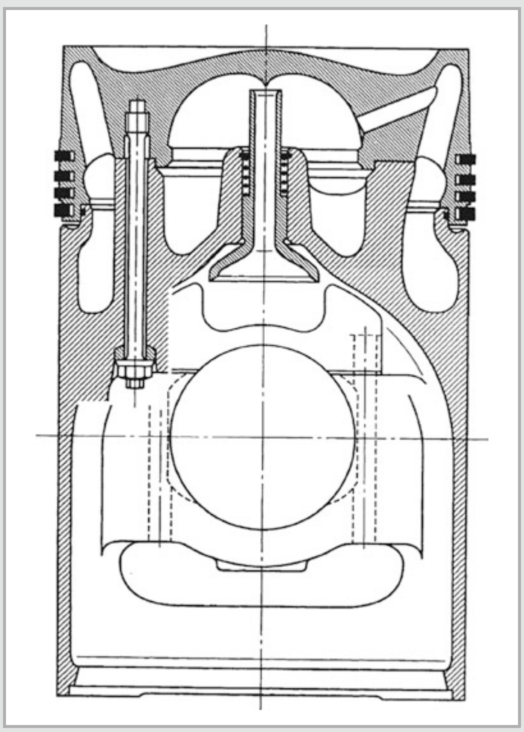

Fig. 18-25 Composite stepped piston with steel top part and modular cast iron skirt for very high cylinder pressures (source: MAN Diesel)

All piston rings are placed in the steel top.

The development of chrome-ceramic coated rings, i.e. chrome rings with ceramic inclusions, combined the high stability of plasma rings with the low wear of chrome rings. Thus, low rates of wear in an order of magnitude of 0.01–0.02 mm/1,000 h were attainable even for the poorest fuels (Fig. 18-26). Hence, modern medium speed diesel engines have chrome-ceramic coated compression rings and chrome coated second and, where applicable, third rings, thus making the ring package highly stable.

Cylinder liner. Separate, single vertical cylinder jackets that hold the cylinder liners provide advantages primarily for larger engines since they reduce actions of adjacent cylinders or ship deformations and thus optimize the roundness of cylinder liners during operation. The water flow and intensive cooling are limited to the upper region of the cylinder liner since they are only required there. The goal is a uniform temperature distribution over the entire surface of the liner to prevent cold corrosion and ensure good lubrication conditions. Together with the stable cylinder geometry, this establishes the prerequisites for low lubricating oil consumption, which should not exceed 0.5–1 g/kWh in modern medium speed engines.

The introduction of piston top land rings in the 1990s heralded a significant advance. Also called anti-polishing rings among other names, they have become widespread. In addition to the cooled piston top land design pictured in Fig. 18-27, uncooled and indirectly cooled designs are employed in which a relative thin-walled ring is inserted directly in the cylinder liner. Ring diameters somewhat smaller than the cylinder liner's actual cylinder bore surface are common to all these designs. In combination with a stepped piston, this effectively prevents "bore polishing" (bare spots caused by hard coke deposits on the piston crown or pitting in the center of the cylinder bore surface). Thus, the cylinder liner, piston top land ring and stepped piston attain a service life of up to 80,000 hours while oil consumption is low. This also includes the low wear values presented in Fig. 18-28 of approximately 0.01 mm/1,000 h attained in conjunction with chrome-ceramic coated rings and measured at the uppermost piston ring's reversal point. The greatest wear, known as bore wear usually occurs at this spot on the cylinder liner.

Cylinder head with valves. As loads increase, nodular cast iron is increasingly being employed for cylinder heads. By virtue of its significantly higher mechanical strength than laminar gray cast iron and in conjunction with a design that facilitates loading, it significantly contributes to the operational reliability of this highly mechanically and thermally loaded component.

Four valves are employed in more highly loaded medium speed engines (Fig. 18-29). Valve cages are increasingly being dispensed with even in larger engines [18-9]. Operational

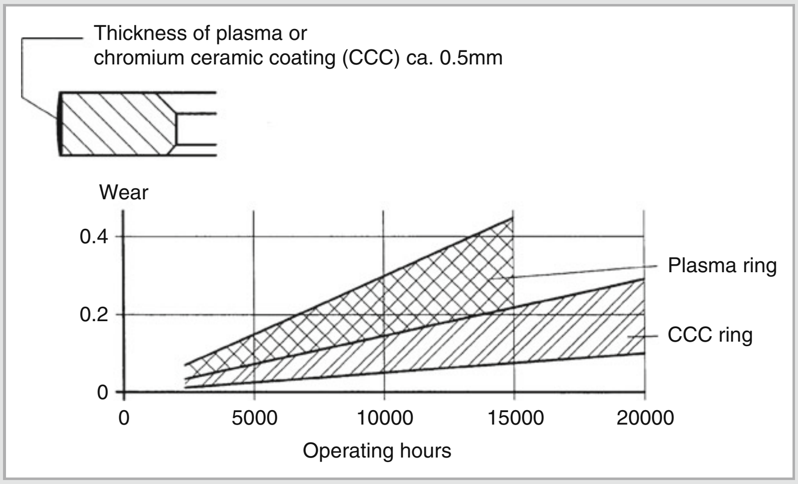

Fig. 18-26
Average wear of the first piston ring in medium speed four-stroke engines operated with heavy fuel oil

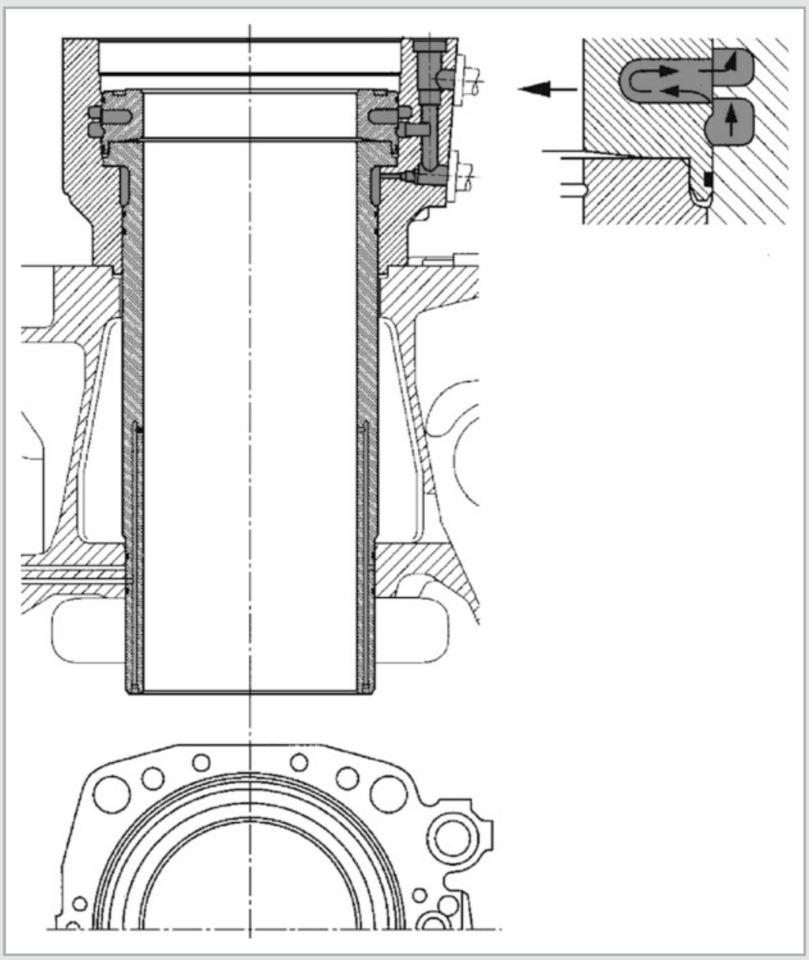

Fig. 18-27
Cylinder liner with water jacket and piston top land ring with bore cooling

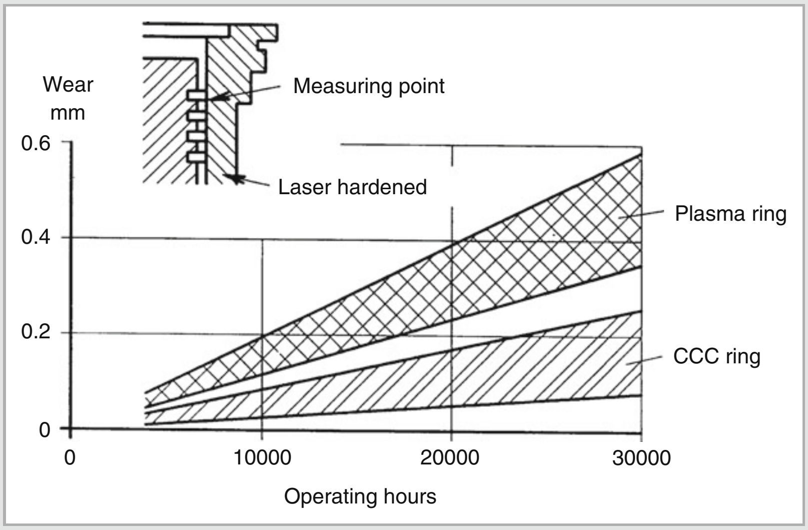

Fig. 18-28
Average wear of the cylinder liner in medium speed four-stroke engines operated with heavy fuel oil

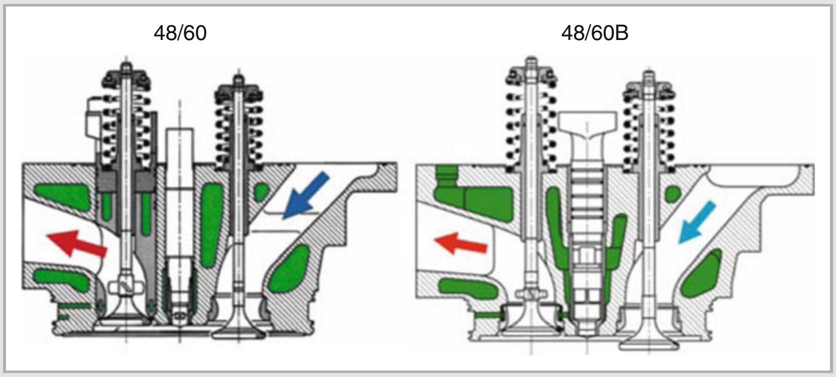

Fig. 18-29
Comparison of the MAN 48/60 engine's cylinder heads with vane valves on the exhaust side (48/60 with exhaust valve cages / 48/60B without exhaust valve cages)

reliability has been increased and valve service lives have been extended so long that a cylinder head must be disassembled for other maintenance work too (e.g. piston rings). This eliminates the maintenance advantage of the valve cage and the disadvantages predominate, e.g. structural complexity, reduced cylinder head stiffness and additional potential leak points. The valve seat inserts are often cooled, at least on the exhaust side.

Both the valve cone and the seat inserts usually have a hard seat facing, e.g. stellites (carbide metals), which assures high wear resistance and prevents pocketing by combustion particulates in heavy fuel oil operation and thus burnouts caused by inadequate valve seat sealing and the resultant discharge of heated combustion gases. Nimonic valves with and without hard seat facing are also used on various occasions.

Practice has demonstrated that valve rotation is absolutely essential during operation when heavy fuel oil is burned. Valves may be rotated by mechanical rotators, e.g. rotocaps. Individual manufacturers employ rotating vanes in the valve stem on the exhaust side. Significantly more intense rotation is obtained from discharging exhaust than mechanical rotators. In the process, the mass inertia of the valve causes rubbing seating on the seat.

18.3.3.4 Injection System

Readers are referred to Sect. 5 for the basics. Limitations of space only permit briefly touching on common configurations and designs for medium speed engines.

Since not only the ratio p_{Zmax}/p_e but also the duration of combustion influences fuel consumption considerably,

attempts have also been made to optimize the duration of combustion in medium speed engines and thus keep it as brief as possible. A relatively close relationship between the duration of combustion and the duration of injection exists in injection systems used solely for direct injection: A brief duration of combustion also requires a correspondingly brief duration of injection. Thus, many manufacturers are now employing high intensity injection systems. Ultimately, this results in comparatively high pressures in the injection system, which in turn has to be appropriately factored into the design of components. In virtually every design, the injection valve is centered in the cylinder head and a multi-hole nozzle conveys the fuel to the combustion chamber.

The ability to adjust injection to the sometimes widely differing combustion characteristics is advantageous when heavy fuel oils are burned. Thus, for example, there are designs that aim to positively influence the combustion cycle by a low pilot injected fuel quantity. Other manufacturers have created the option of influencing the ignition point or injection timing to be able to appropriately respond to different ignition delays.

In conjunction with the existing statutory limits on emissions, such measures will increasingly gain importance since this method, among others, can influence NO_x emission by shifting the ignition point.

Given their variability, common rail systems furnish engine developers a broad range and higher flexibility of injection parameters. In addition to variable start of injection and injection pressure, multiple injections may be necessary for optimal combustion with low pollutant content.

Common rail systems are increasingly also being implemented in medium speed engines despite the problems that arise in heavy fuel oil operation from utilizing heavy fuel oils with a viscosity of up to 700 cSt (at 50°C) since these fuels must be preheated to a temperature of up to 150°C to obtain the requisite injection viscosity. These problems are compounded by the high content of abrasive particulates and aggressive constituents present in heavy fuel oils. Injection components must function reliably at high temperatures under these operating conditions.

A pressure accumulator (rail) extending along the entire length of the engine is problematic for large diesel engines because of thermal expansion and the options to manufacture such a component for 1,600 bar with radial bores. Hence, the pressure accumulators are divided into several segments in the systems introduced so far by Wärtsilä and MAN Diesel [18-10, 18-11]. The fuel supply can also be spread to several high pressure pumps. Supplying high pressure fuel to the accumulator system through two or more high pressure pumps has the additional advantage of enabling engine operation even when one of the pumps fails.

Based on concepts with a segmented rail, MAN Diesel developed a modular system for several engine types (Fig. 18-30).

Along with the advantage of greater flexibility to adapt to different numbers of cylinders and better utilization of

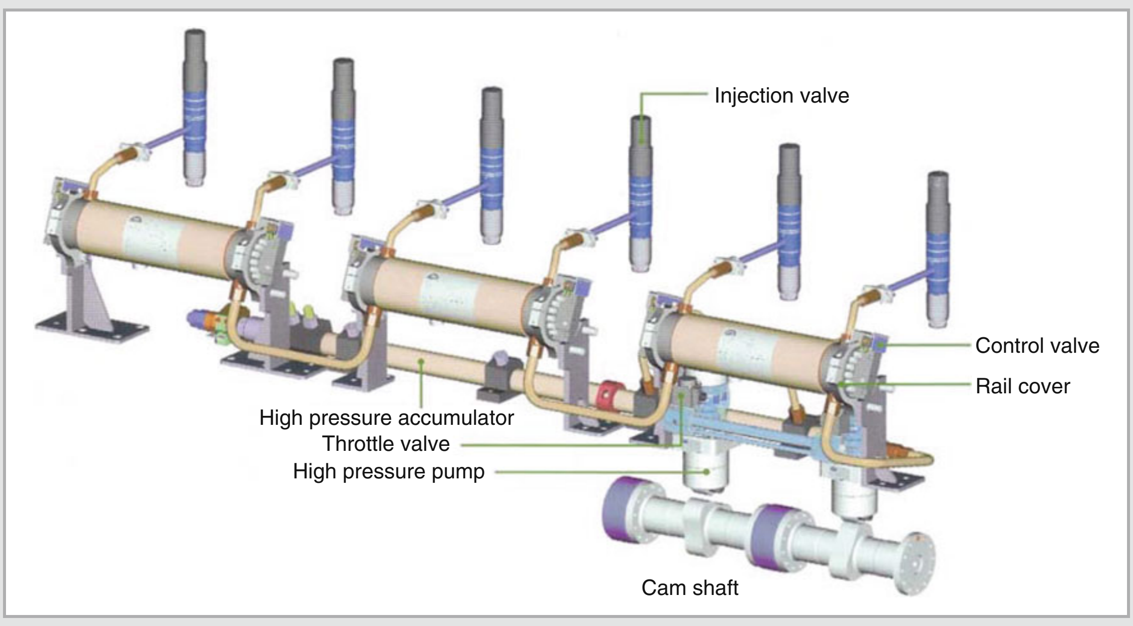

Fig. 18-30 Configuration of MAN Diesel's common rail system

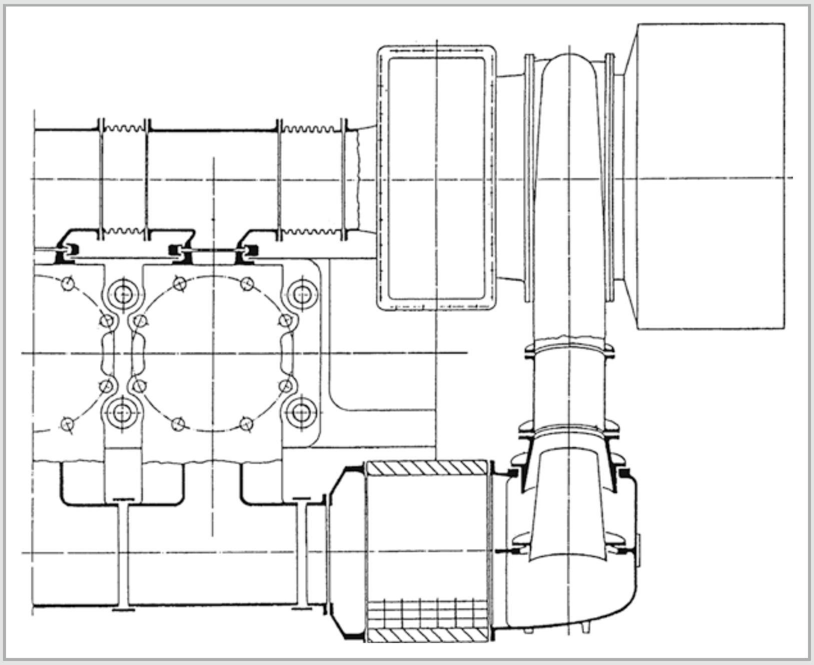

Fig. 18-31
Constant pressure supercharging system with flow-optimized ports and diffusers between the cylinder head and exhaust manifold and after the compressor for pressure recovery

existing space by compact units, segmentation into individual rail modules provides further advantages for assembly and the stocking of spare parts [18-12].

The common rail system is expected to supersede classical mechanical injection with a single plunger pump driven by a camshaft in the future.

18.3.3.5 Supercharging System

Readers interested in the theory of supercharging are referred to Sect. 2.2. This section touches on different aspects of design and implementation, which are typical for medium speed engines.

Modern medium speed engines are almost exclusively equipped with exhaust gas turbocharging. Axial or radial turbochargers are employed depending on the engine size. The advances achieved in turbocharger engineering in recent years now allow a pressure ratio of five and above in one stage. A further increase is in development.

Both pulse and constant pressure supercharging are employed in engines. Constant pressure supercharging has become increasingly established for medium speed engines in recent years since the advantages, e.g. lower fuel consumption, uniform turbine pressurization, simpler exhaust manifold configuration and no numbers of cylinders disadvantaged by supercharging, outweigh the disadvantage of poorer accelerating performance. This can be mitigated by narrow exhaust manifolds and – when necessary – harmonized with the conditions in pulse operation by appropriate additional measures (e.g. Jet Assist in which the compressor is briefly pressurized with compressed air during the phase of revving up).

Designing the ports that conduct air and exhaust to facilitate the flow and correctly configuring the diffusers for pressure recovery beneficially influences the efficiency of the entire supercharging system. This has a positive effect on fuel consumption (Fig. 18-31).

One problem with one-stage supercharging in conjunction with high brake mean effective pressures is the increasing difficultly to optimally cover an engine's air requirement throughout the entire load range because of the different characteristic curves of the engine and turbocharger. However, thoroughly satisfying results can be obtained by such measures as recirculating charge air in the lower load range and, if necessary, discharging exhaust gas at excess load (wastegate).

Infinitely variable turbine geometry would be the optimal solution. However, tests with adjustable guide vanes resulted in considerable problems in heavy fuel oil operation caused by contamination.

18.3.4 Operational Monitoring and Maintenance

In addition to electronically assisted control, timing and monitoring systems, which serve to optimize operation,

diagnostic and trend systems are also employed. Thus, an operator receives information on the system's current status, which is intended to facilitate decisions on measures to be taken.

The latest state of development is the use of expert systems that not only display a system's instantaneous status but also quite specifically inform the operator which component has to be serviced or replaced because of changed engine operating characteristics. This is allowing a switch from scheduled to condition-based maintenance [18-13].

Development in recent years has made it now possible to plan long maintenance intervals for components of modern medium speed engines despite increased loads. Good accessibility and the use of appropriately designed special tools make the work performed on wear parts, e.g. piston rings, injection nozzles, intake and exhaust valves, main and connecting rod bearings, etc., substantially easier. Potential errors during the performance of maintenance work have largely been minimized, a fact that contributes considerably to an engine system's operational reliability.

18.3.5 Exhaust Emission

Measures that improve exhaust emission in medium speed four-stroke diesel engines are primarily aimed at reducing nitrogen oxides NO_x and soot production during combustion. The latter not only causes exhaust blackening but is also responsible for the emission of particulate matter (see Sect. 15.3). This is exacerbated by the operation of large diesel engines predominantly with sulfurous heavy fuel oils (see Sect. 4.3.4.2). In addition, increased soot formation, made noticeable by heavy plumes of smoke, particularly occurs at light load. This is a problem for seagoing ships when they are maneuvering in ports.

In-engine measures are initially an expedient remedy, e.g. improved injection, modified valve timing, etc. [18-8, 18-14, 18-15]. External measures, e.g. the use of water/fuel emulsions or particulate filters (see Sect. 15.5), generally complicate the engine system's handling and increase its susceptibility to faults.

Following this principle, MaK's first step during its development of low-emission large diesel engines was to reduce NO_x emission in compliance with IMO specifications (see Sect. 15.2.3.3). Then, smoke emission was lowered below the visibility limit. The long-term goal was a low-emission engine (LEE) with an eye toward future requirements [18-16].

It turned out that several measures have to be combined as a function of load. The Miller cycle (see Sect. 2.2.4) reduces the maximum combustion temperature in the upper load range and thus the formation of NO_x. The charge loss connected with this can be compensated by increased boost pressure, provided the limits of one-stage supercharging are not reached (see Sect. 2.2.3). A larger compression ratio in conjunction with a longer stroke ($s/D = 1.5$) can reduce NO_x emissions significantly but often with heavier smoke emission at light load. However, the use of flexible camshaft technology (FCT) at lower power can keep exhaust blackening below the visibility limit (smoke number $SN \leq 0.4 \ldots 0.5$; see Sect. 15.6 and Fig. 18-32). To do so, the injection cams' rated power is advanced at light load upwards of approximately 25% power so that, in conjunction with a modified injection pump, atomization is improved and combustion is low in soot. At the same time, the intake valve opens and closes later, thus dispensing with the Miller effect, while the exhaust valve opens earlier to increase the boost pressure by a larger exhaust gradient.

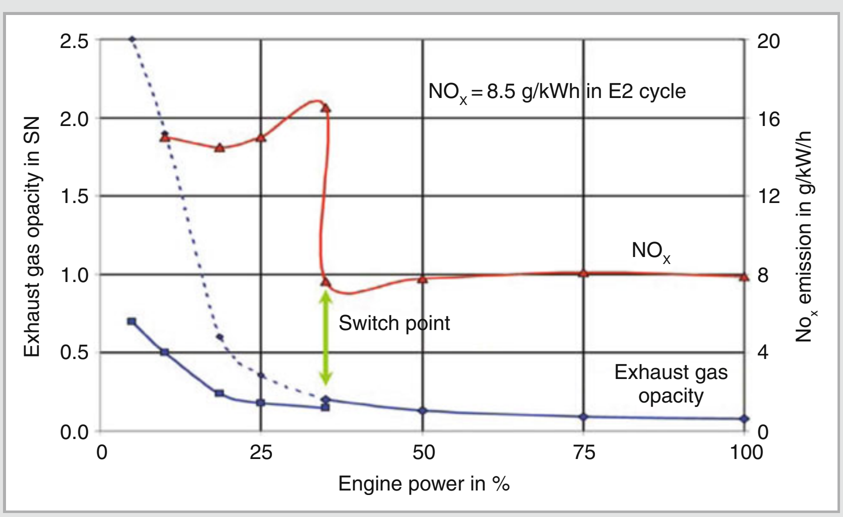

Fig. 18-32
NO_X emission and exhaust gas opacity (SN) of an exhaust-optimized Caterpillar M 43 C marine diesel engine (current IMO limit: NO_X = 12.9 g/kWh, SN: smoke number based on the Bosch filter method)

18.3.6 Implemented Engines

Given the abundance of different medium speed engines from the widest variety of manufacturers (the diversity of types is especially great in the lower power range), only a few examples can be singled out here.

Figure 18-33 pictures MAN Diesel's family of large medium speed engines, consisting of the L58/64, L48/60, L40/54 and L32/40 models with cylinder outputs of 1,400, 1,200, 720 and 500 kW at speeds of 428–750 rpm. The standardized engineering of these four engines, each of which is

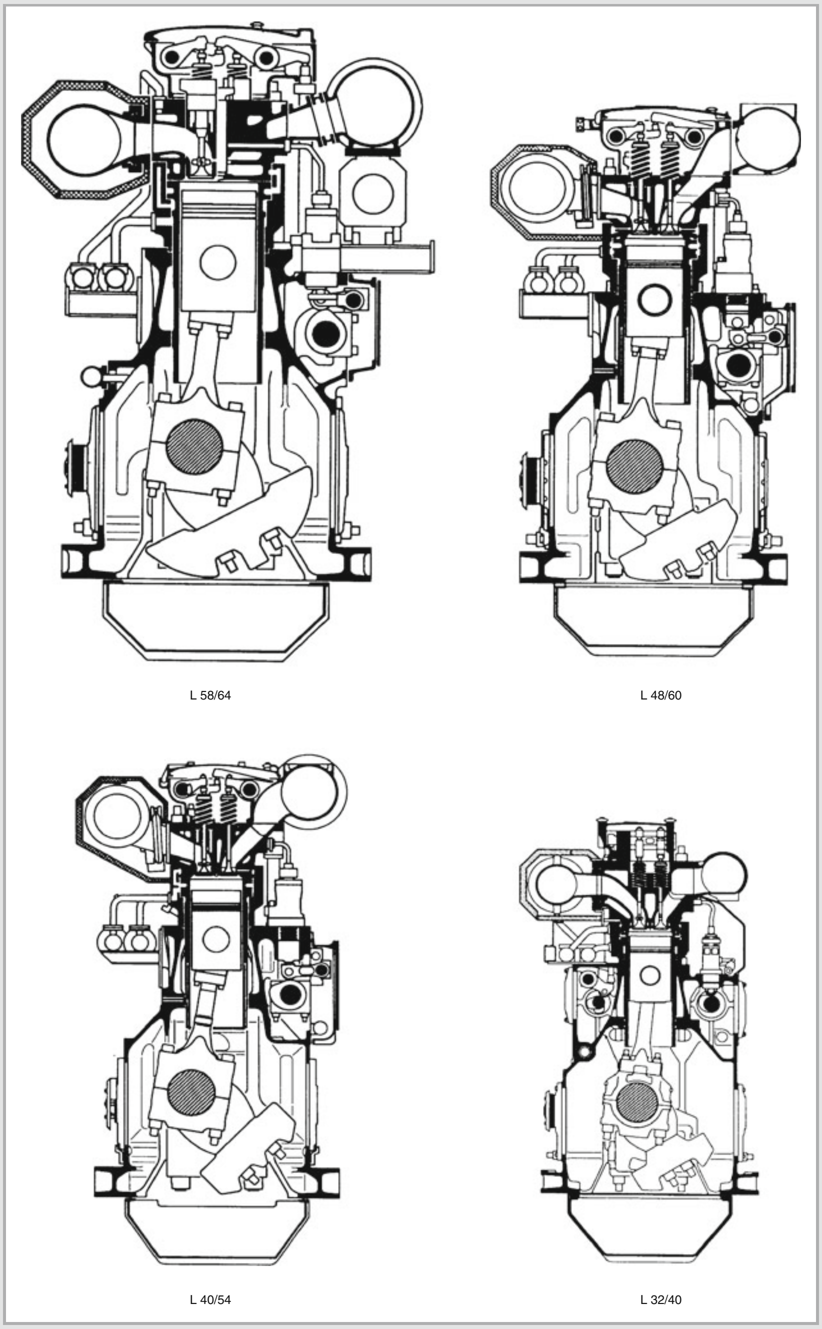

Fig. 18-33
Upper power range of MAN Diesel's medium speed engine family, consisting of four inline engines of largely identical design

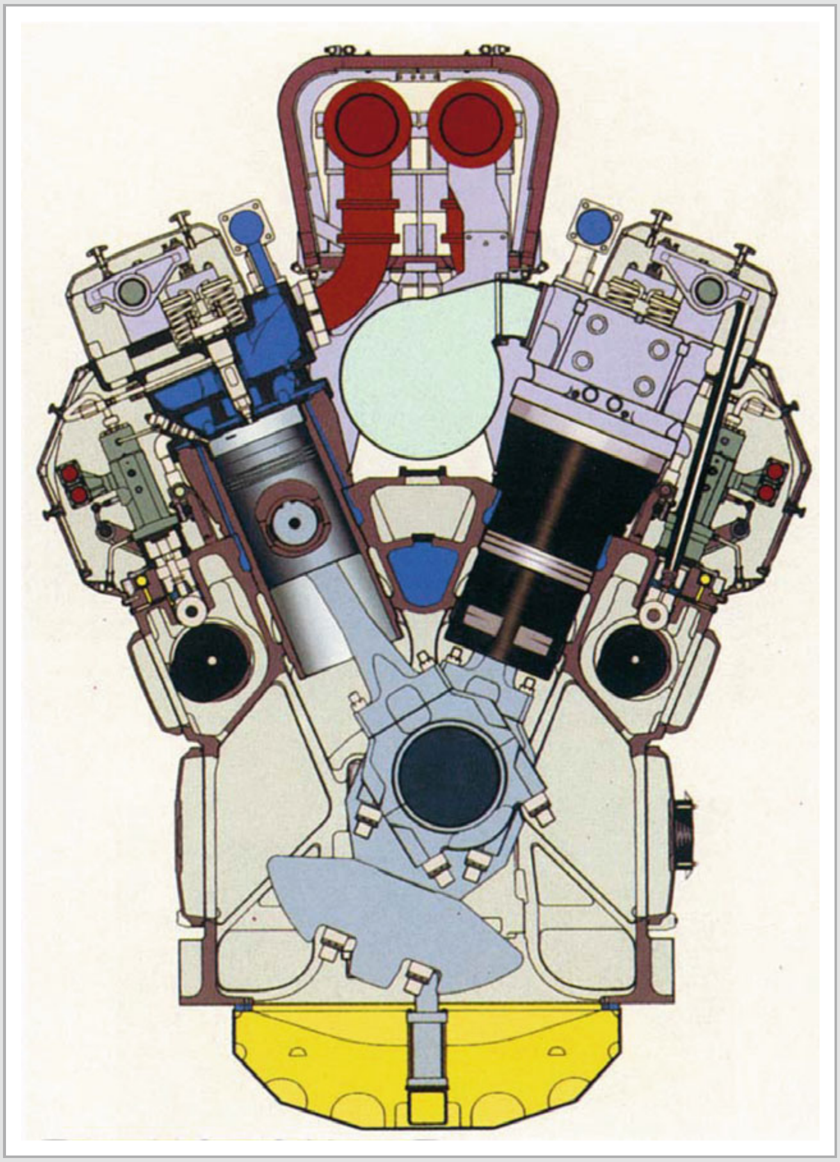

Fig. 18-34
Wärtsilä 46

constructed in an inline configuration, is remarkable. In addition to the type with 320 and 480 mm, a V version is also available.

A broad range of medium speed engines in the bore range of 250–350 mm is commercially available. Among others, the Wärtsilä 32 has been very successful in recent years. With a cylinder diameter of 320 mm and a stroke of 400 mm, a cylinder output of 500 kW is reached at a speed of 750 rpm. This engine is also built in an inline and V design with six, seven, eight and nine or twelve, sixteen and eighteen cylinders.

Pictured in cross section in Fig. 18-34, the larger Wärtsilä 46 has a cylinder diameter of 460 mm and a stroke of 580 mm. It is available with cylinder outputs of 975, 1,050 and 1,155 kW at 500 rpm. For the W46F stage of development, the speed was increased to 600 rpm and thus a cylinder output of 1,250 kW was attained.

MaK's M20 engine is representative of smaller dimensions in the range of a cylinder bore of 200 mm (Fig. 18-35). The engine has a number of design features that had been reserved for larger medium speed engines, e.g. individually attached

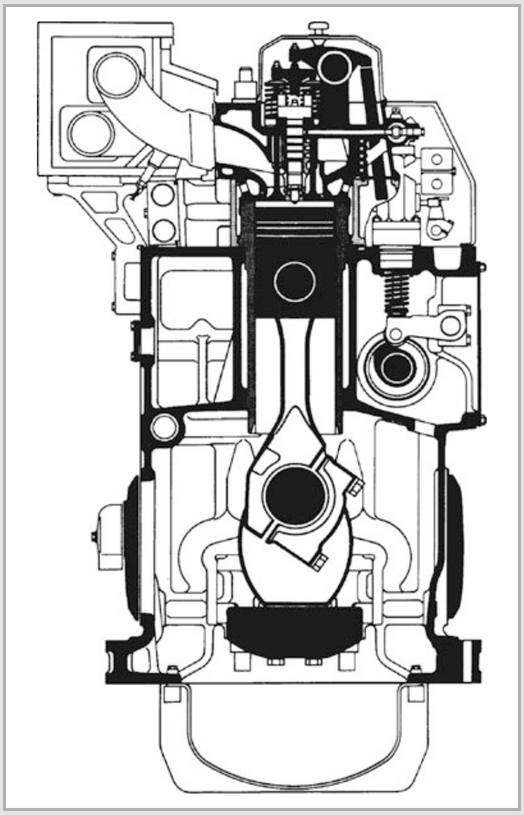

Fig. 18-35 Cross section of MaK's M20 engine

cylinder jackets. With a bore of 200 mm, stroke of 300 mm and a speed of 1,000 rpm, the engine presently available in an inline version with six, eight and nine cylinders has a cylinder output of 190 kW.

18.3.7 Outlook

A number of requirements are imposed on medium speed four-stroke diesel engines and will be in the future too. Naturally, demands for higher cost effectiveness, reliability and simplicity of maintenance are foremost from the perspective of operators. The technical concepts of modern medium speed four-stroke diesel engines largely meet these demands. High service life primarily means low wear values for the most important components. Simplicity of maintenance, i.e. the use of easily handled hydraulic tools and good accessibility of serviced components are part of cost effectiveness.

Of course, cost effectiveness also means low specific consumption of fuel and lubricating oil. Elevation of the maximum cylinder pressure, optimization of the combustion process and developments in supercharging technology have made it possible to lower fuel consumption substantially in recent years. Today, a medium speed four-stroke engine is able to convert more than 50% of the fuel-based energy into mechanical work.

Along with a further concentration of power, the reduction of emissions will be the priority as medium speed four-stroke engines are refined in the coming years (see Sect. 18.3.5 and Part IV of this book).

In addition to reducing NO_x emission, intensive work is also being done on further lessening particulate emission. The goal for medium speed engines is to produce invisible exhaust from idle to full load by suppressing soot production, even in heavy fuel oil operation.

In the ideal case, this would prevent sulfurous combustion products from binding to soot and increasing particulate emission. However, as mentioned in Sect. 4.3, low exhaust blackening is not a criterion for equally low particulate emission when sulfurous fuel is burned.

Elastic and semi-elastic installations in ships, which reduce structure-borne noise, will continue to grow in importance for noise emission. In addition to noise reducing and noise absorbing measures at the engine itself, which have largely been exhausted, appropriately sound engineered machine rooms or, where feasible at these engines' dimensions, encapsulation will reduce noise even further.

18.4 Two-Stroke Low Speed Diesel Engines

18.4.1 Development and Features of Two-Stroke Low Speed Diesel Engines

18.4.1.1 Development of Two-Stroke Low Speed Engines

Soon after the introduction of the first diesel engine with the four-stroke principle envisioned by Diesel, Hugo Güldner proposed and designed – against Diesel's counsel – a two-stroke diesel engine in 1899, which was denied success though [18-17]. The Sulzer bothers in Winterthur deserve the credit for introducing the first operable two-stroke diesel engine implemented as a marine engine in 1906. Other firms such as MAN-Nürnberg, Krupp-Germania-Werft, Burmeister & Wain (Copenhagen) soon followed. Together with larger cylinder dimensions, the two-stroke principle was viewed as an opportunity for the marine engine to compete against reciprocating piston steam engines with their large energy units.

While a two-stroke engine with its two working strokes was theoretically able to produce twice the power of an equally large four-stroke engine, the real increase was only approximately 60% because of losses due to the lower purity of the charge and the required compression of the scavenging air.

This has led to a great diversity of engine concepts over decades, which, on the one hand, are characterized by generally similar basic features and, on the other hand, also bear features specific to manufacturers, e.g. Doxford, Grandi

Motori Trieste (formerly Fiat), Götaverken, Stork, Werkspoor, etc., and are characterized by the following criteria:

Working principle:
– single-acting piston,
– double-acting piston and
– opposed piston.

Methods of scavenging:
– uniflow scavenging,
– loop scavenging and
– cross flow scavenging.

Methods of supercharging:
– mechanical supercharging,
– exhaust gas turbocharging and
– combined mechanical supercharging and exhaust gas turbocharging.

Combinations of the individual principles yielded the widest variety of designs, some of which were able to hold their own into the 1970s. All manufacturers worked with direct fuel injection instead of air injection, which had been common at first.

On the one hand, the two "oil crises" in the second half of the 1970s and the early 1980s once again triggered great strides in development in terms of diesel engines' fuel economy. On the other hand, they set off a concentration of the only three firms now remaining worldwide, which develop and engineer two-stroke low speed diesel engines and, in addition to licensing them, also partly manufacture them themselves. Since the mid 1990s, these (Fig. 18-36) have been:
– MAN Diesel SE (formerly MAN B&W Diesel AG),
– Wärtsilä (formerly Gebrüder Sulzer/New Sulzer Diesel) and
– Mitsubishi Heavy Industries (MHI).

All three suppliers are pursuing the same concept: a low speed single-action, exhaust gas turbocharged, uniflow scavenged two-stroke diesel engine.

Thus, the once great diversity of concepts has essentially given way to one concept that appears quite logical today.

18.4.1.2 The Transition to Uniflow Scavenging

Until the mid 1970s, stroke to bore ratios among all the still commercially active manufacturers fluctuated between 1.7 and 2.1. These allowed loop or cross flow scavenging without losses of scavenging efficiency. The distinctive feature of these two scavenging systems was their particular simplicity of design, which functioned without an exhaust valve in the cylinder cover. This made maintenance exceptionally simple and user-friendly, which helped this engine type to particular commercial success in the 1960s and 1970s.

However, the first of the oil crises (1973) subsequently triggered a clear turning point in development. The astronomical increase of the share of fuel costs in total operating costs after 1973 was crucial to this. Thus, not only was the maximum cylinder pressure increased in quick succession to save fuel but developments in the direction of fuel economy also commenced on the shipbuilding side. Propeller speeds and propeller diameters grew smaller and thus propeller efficiencies larger.

This inevitably resulted in larger stroke to bore ratios for two-stroke engines in order to be able to maintain the mean piston velocity and thus the power output.

MAN-Augsburg and Sulzer were able to keep pace with this development with their simple valveless engines with a stroke to bore ratio of approximately 2.1 until the end of the 1970s. Then, however, market demand for even lower speeds forced a transition to uniflow scavenging. MAN solved this problem in early 1980 by taking over the diesel operations of the Danish firm Burmeister & Wain (B&W) since large B&W two-stroke engines with a stroke to bore ratio of approximately 2.4 had already been operating with uniflow scavenging for a long time.

Combining the proverbial reliability of valveless, loop scavenged engines with the longer strokes being demanded from uniflow scavenged engines subsequently became essential to development and design. Introducing its (super-long stroke) RTA series [18-18] in the early 1980s, Sulzer mastered this challenge first by adopting uniflow scavenging with a central exhaust valve and increasing the stroke to bore ratio to approximately 3.0 for the first time. This made substantially lower rated speeds possible (67 rpm in the largest engines).

Such engine models were available from MAN B&W (L. . .MC/MCE) and Mitsubishi (UEC. . .L) soon afterward. Determinative for operating costs, these extremely long-stroke engines with concentrated power reduce the fuel consumption of marine systems not only by their better combustion process but also by their better propulsion efficiency.

Today, none of the three manufacturers' advanced two-stroke low speed engines differ from one another in terms of their basic principle.

Observable in other industrial products at the close of the twentieth century, "natural" selection had occurred here too, not least because of the advanced design and computational tools that had become available and permit selecting the logically correct solution rapidly and efficiently. Despite the strongly increased specific powers, lower engine weights and related higher load in recent years, the reliability of these machines has been improved to the point that overhaul intervals of three years are now feasible. Today, the basic difference between the three manufacturers is their different concepts for electronically controlled injection.

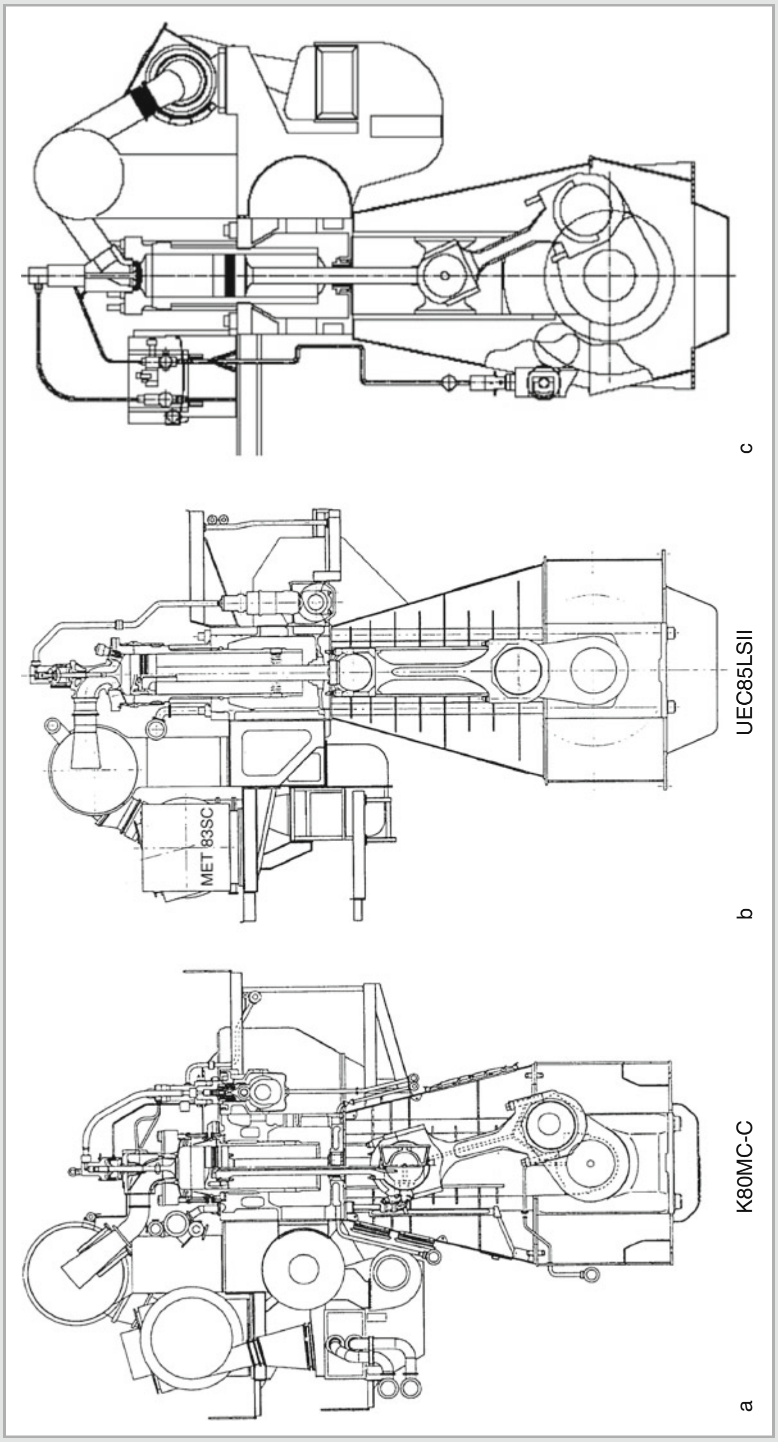

Fig. 18-36
Modern designs of two-stroke low speed diesel engines with uniflow scavenging: (**a**) MAN B&W: S90MC-C (D = 900 mm); (**b**) Mitsubishi: UEC85LsII (D = 850 mm); (**c**) Wärtsilä RT-flex82C (D = 820 mm)

18.4.1.3 Features of Modern Two-Stroke Low Speed Engines

In its more than ninety year history, the low speed diesel engine has experienced tremendous technical development made possible by:
– ongoing conceptual development,
– the transition to uniflow scavenging,
– advances in supercharging equipment and turbocharger design,
– new knowledge in materials technology,
– the utilization of advanced theoretical and experimental methods of development and
– the transition to electronically controlled injection and valve timing.

The *brake mean effective pressure* p_e of approximately 5 bar in the naturally aspirated engines of the early 1950s have now risen to 20 bar in supercharged engines. This corresponds to an increase of *effective brake work* w_e from 0.5 to approximately 2.0 kJ/dm^3. *Mean piston velocity* has increased from approximately 5 to over 9 m/s.

In the process, the *maximum cylinder pressure* $p_{Z\max}$ has increased from approximately 50 bar to more than 160 bar. Pronounced *long strokes* with stroke to bore ratios of between approximately 3.0–4.2 may be regarded as the typical design feature of today's large two-stroke diesel engines. Their cylinder diameter D can be between 260 and 980 mm.

Uniflow scavenging with an actively controlled exhaust valve is now implemented almost exclusively as the scavenging process.

Characterized by the *specific power per unit piston area* P_A, the power density reaches values of more than 790 W/cm^2. Thus even large two-stroke low speed engines constitute a demonstrably "high-tech" product (see Sect. 1.2).

At the same time, in conjunction with turbocompounding and exhaust gas heat recovery, *specific fuel consumption* has dropped from approximately 220 to 156 g/kWh. This corresponds to an effective efficiency of approximately 55%. Thanks to the ruggedness of the basic concept, a two-stroke low speed diesel engine is now also able to burn the poorest grades of fuels.

During this period, the cylinder outputs of the largest engines rose from a few 100 kW to over 5700 kW. This has made power outputs of over 80,000 kW possible with one engine. The trend toward ever larger container ships still only equipped with one propeller and expected to operate at identical or even higher speed while allowing for increased time requirements for loading and unloading as well as liner traffic has already given rise to demands for power outputs of 100,000 kW and more in recent years.

According to Eq. (1-13), a linear correlation of specific work w_e (or brake mean effective pressure p_e), mean piston velocity c_m and the number of cylinders z exists for engine power. However, engine power increases with the square of piston diameter D. Logically, manufacturers have so far attempted to meet demands for greater power by a steady, incremental increase of specific power data as well as with engines with larger piston diameters. The upper limit is now at $D \leq 1$ m but a further increase of D at an appropriately designed thermal load (see Sect. 7.1), component mass, mixture formation and combustion is definitely conceivable. Increasing the numbers of cylinders above the formerly common maximum number of twelve remains another option.

Wärtsilä already offers large two-stroke engines with a piston diameter of 960 mm (RTA96C and RT-flex96C) and up to $z = 14$ cylinders inline, thus, obtaining a cylinder output of 5,720 kW and a total engine power of 80,080 kW at values of $p_e = 18.6$ bar or $w_e = 1.86$ kJ/dm^3 and $c_m = 8.5$ m/s. This engine type is the largest and most powerful diesel engine ever built. The first engines of this type were commissioned in 2006.

Higher numbers of cylinders than $z = 14$ can hardly be considered realistic at present since engine mass and overall length inevitably increase as z increases. This particularly raises problems with the engine foundation and the load of a ship's hull.

For this reason, MAN has been carrying an even larger engine, the K108MC-C, in its program since 2002. With a nominal cylinder output of 6,950 kW, this engine would achieve a total power of 83,400 kW with a maximum number of cylinders of $z = 12$.

Further considerations to reduce overall length and engine mass have led to the option of a V configuration of the cylinders. Relevant designs and simulations by MAN revealed that the engine mass of a twelve cylinder V engine with a piston diameter of 900 mm would be reduced by 15% and the length by 6.8 m, i.e. by approximately 30%. It also turned out that no major problems in terms of maintenance and accessibility are to be expected. However, none of these two engine variants with a 108 cm bore or V configuration has been produced so far [18-19].

Figure 18-37 charts the chronological development of the important engine parameters of Wärtsilä's large two-stroke diesel engines.

18.4.2 Modern Two-Stroke Low Speed Engine Design

18.4.2.1 Engine Families, Power Map

All commercially available two-stroke low speed engines have the same concept and thus similar design features. Thus, a detailed description of one manufacturer's engine is sufficient.

Since engine power and speed are firmly interrelated for a direct drive propeller, a closely graduated engine family of differing bores and stroke to bore ratios is a necessity for every manufacturer. Figure 18-38 presents partly overlapping maps of one manufacturer's offerings as an example. This makes it possible to select the optimal engine while considering criteria such as installation dimensions, number of cylinders, fuel consumption, etc.

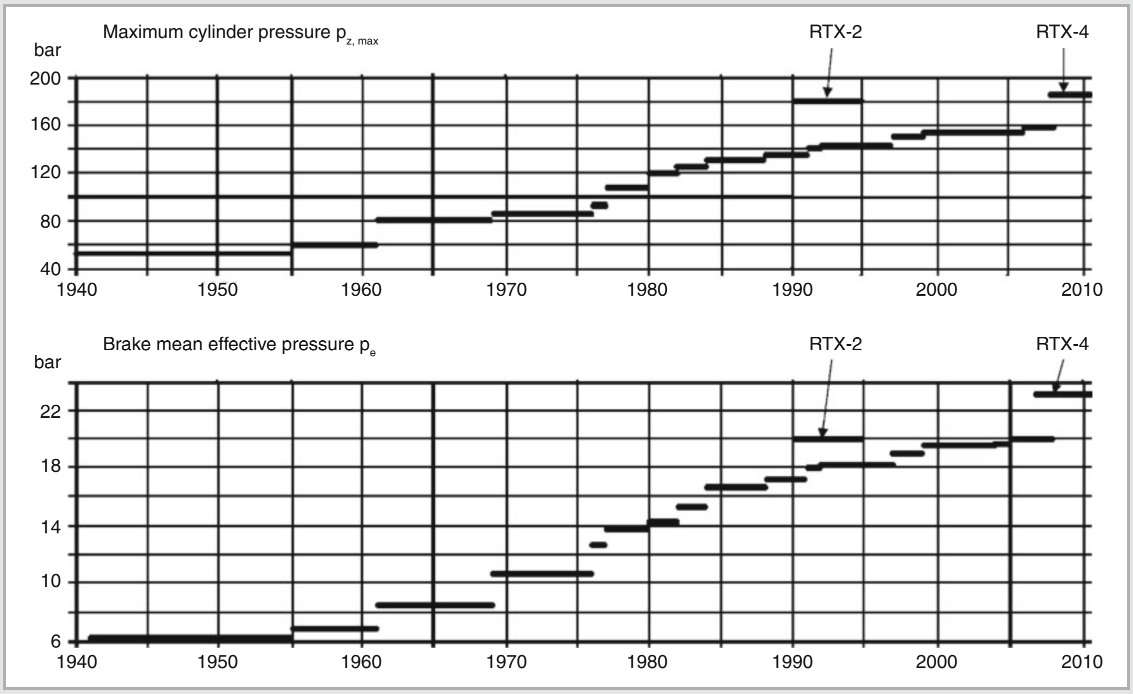

Fig. 18-37 Development of the engine parameters of maximum cylinder pressure $p_{Z,max}$ and brake mean effective pressure p_e over the last 65 years (Wärtsilä RTX-2 and RTX-4 research engines)

Different engines in the Wärtsilä family described here include:

– engines with stroke to bore ratios of up to approximately 3.5 (RTA52U, RT-flex60C, RTA62U, RTA72U, RTA82C/RT-flex82C,RTA96C/RT-flex96C), which, produce high powers at relatively high speed, above all for faster ships, e.g. container ships or car carriers, and
– engines with stroke to bore ratios higher than 4.0 (RTA48T, RTA58T/RT-flex58T, RTA68/RT-flex68, RT-flex82T), which, at correspondingly lower speeds, are intended for slower ships with larger propellers, e.g. tankers and cargo ships.

Every engine's power map is specified by the vertices R1/R2 to R3/R4 (Fig. 18-38), the engine's rated power being freely selectable within this map for a particular application. Depending on demand, this allows full utilization of the maximum power (point R1) or a variant with reduced power with the advantage of lower consumption and/or lower propeller speed.

The R1+ map concept represents another design variant, which delivers the same propulsion power in conjunction with fuel consumption reduced by 2 g/kWh at increased speed and reduced mean pressure.

18.4.2.2 Engine Design

Engine frame. The 14RT-flex96C (80,080 kW, 102 rpm) is currently the world's most powerful diesel engine in operation. The typical features of a modern two-stroke low speed are explained using the example of the Sulzer RTA96C diesel engine illustrated in Fig. 18-36. The following engine parameters apply to this engine:

stroke/bore (mm/mm)	2,500/960 (= 2.6)
cylinder output (kW)	5,720
rated speed (rpm)	102
brake mean effective pressure (bar)	18.6
specific work (kJ/dm^3)	1.86
mean piston velocity (m/s)	8.5
specific consumption (g/kWh)	171
maximum cylinder pressure (bar)	145
power per unit piston area (W/cm^2)	790

The engine frame consists of a rigid, welded structure with a bedplate and an A-frame in which white metal crosshead guide rails are integrated. In modern engines, the overlying cast cylinder jacket, which holds the cylinder liners, is "dry", i.e. does not contain a cooling water

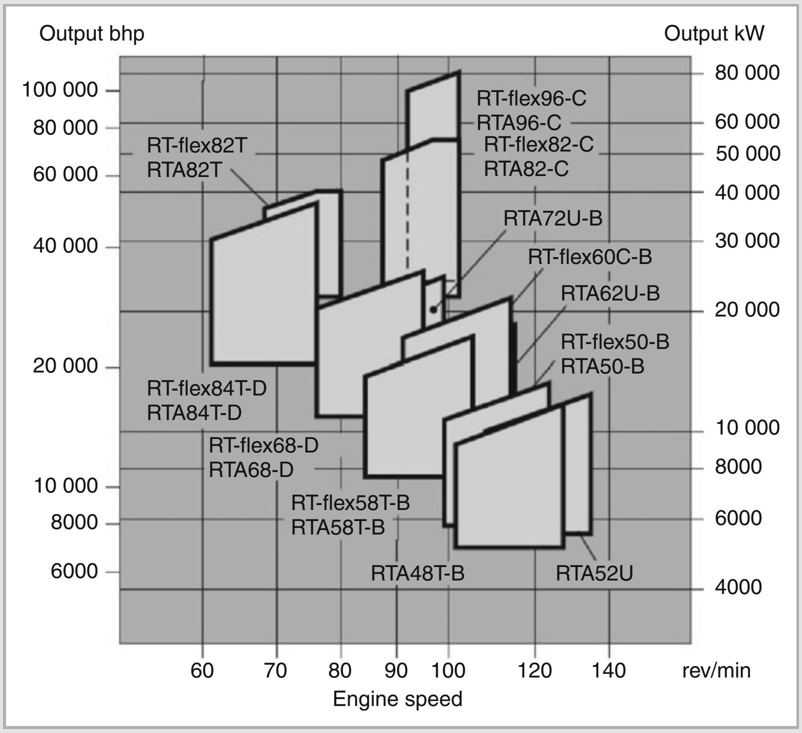

Fig. 18-38
Power and speed maps for Wärtsilä's two-stroke diesel engine program

space. All three components are bolted together and thus provide the stable structure demanded. All the bolted joints are easily accessible from the outside. The stresses and deformations in these important engine components are very low and assure high reliability (Fig. 18-39). This basic structure is similar in all Wärtsila two-stroke engines and its principle is also found among the other manufacturers.

Crankshaft assembly. Crankshaft assembly is the central domain of engine design. The design must ensure that every crankshaft assembly element, e.g. crankshaft, push rod (connecting rod), crosshead, piston rod, bearing etc., functions properly during the entire service life of sometimes more than 25 years.

A crankshaft consists of forged single throws connected with the main bearing journals by a transverse press fit (shrink fit). Apart from shrinking, throws and main bearing journals may also be connected by narrow gap welding. The stiffness and loads of the very slender throws are meticulously optimized by finite element calculations and dynamic measurements (Fig. 18-40). The calculations utilize dynamic analyses of the crankshaft and bearing loads, taking into account the stiffness and damping of the radial and axial bearing structures including the effects of cylinder damping. This makes small distances between cylinder bore center lines (approximately 1.75 x bore) possible with high reliability.

A very short connecting rod is employed to limit engine height despite the large stroke/bore ratio. The push rod ratio r/l (crank radius/connecting rod length) is 0.5–0.45, i.e., approximately twice as large as usual. However, generously sized crosshead shoe surfaces can readily absorb high side forces.

The actual crosshead bearing constitutes a distinctive feature of this type of engine [18-20]: Only it oscillates and its load vector always points downward. This interferes with reliable hydrodynamic lubrication and the supply of oil. One corrective is hydrostatic lubrication with approximately 12 bar oil pressure in oil chambers specially provided for this. They briefly elevate the crosshead journal during every revolution (Fig. 18-41) to assure oil is supplied.

The particularly high reliability of the main bearing, connecting rod bearing and crosshead bearing is a characteristic of this engine design. The following factors contribute to this:

– The specific loads are kept low by appropriate sizing.
– The white metal wear layer has excellent emergency running properties and is very flexible.
– Unique to two-stroke engines, the clear division between the combustion chamber and crank chamber protects these bearings against the effect of combustion products.

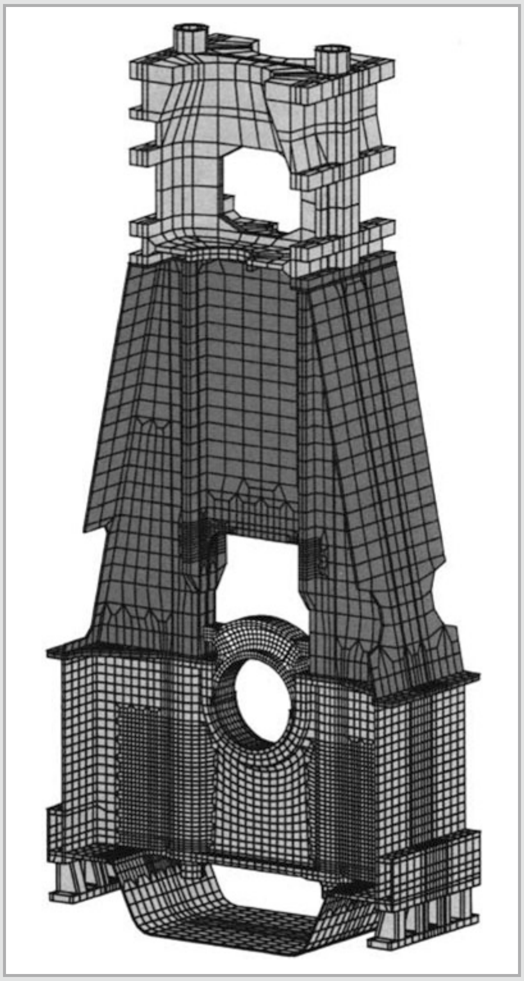

Fig. 18-39 Structural model of an engine frame (RTA96C) consisting of welded bedplate and A-frame with cast single cylinder blocks for FE calculation

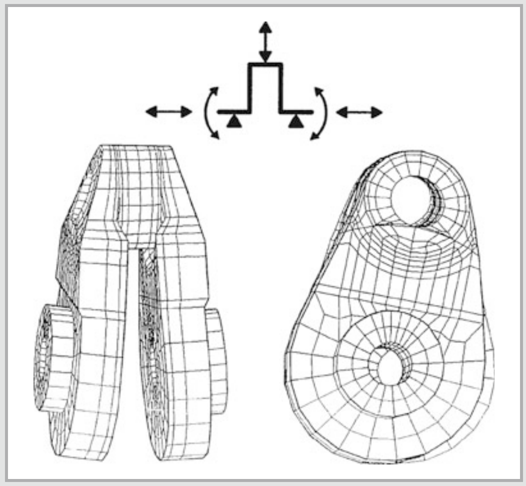

Fig. 18-40 FE calculations and measurements of deformations for a single crankshaft throw

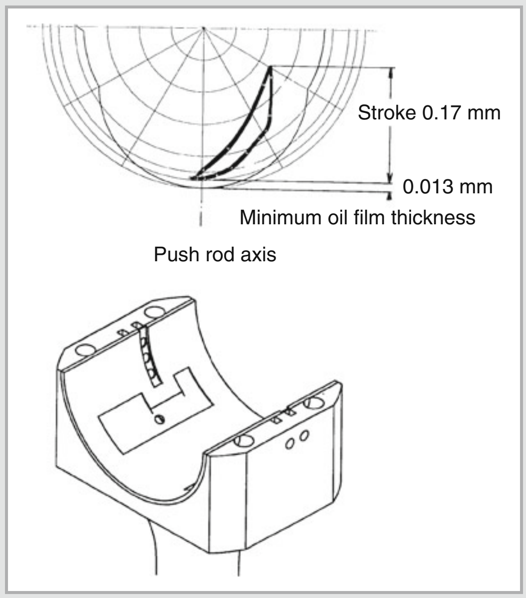

Fig. 18-41 Simulation and measurement of oil film thickness for a crosshead bearing

Combustion Chamber. Since energy (fuel mass) conversion is high when cylinder bores are large, meticulous design of the combustion chamber is the prerequisite for operating reliability. The simultaneous absorption of high thermal and mechanical loads in modern supercharged diesel engines led to the introduction of bore cooling at the end of the 1970s (Fig. 18-42), which Sulzer had already patented at the end of the 1930s. It provides effective, precisely metered cooling of the combustion chamber components (piston, cylinder liner, cylinder cover, valve and valve seat) at simultaneously higher rigidity without the necessity of thermal coatings, so-called cladding. This made it possible to keep the temperatures of the combustion chamber walls within permissible limits despite the steadily increased specific power (Fig. 18-43). Moreover, the water cooling initially employed as piston cooling was replaced in all large two-stroke engines by operationally simpler oil cooling. In particular, spray nozzles (jet shaker) increased the heat transfer in the cooling bores by 50% over the previous "shaker effect". Other engine manufacturers have adopted this principle.

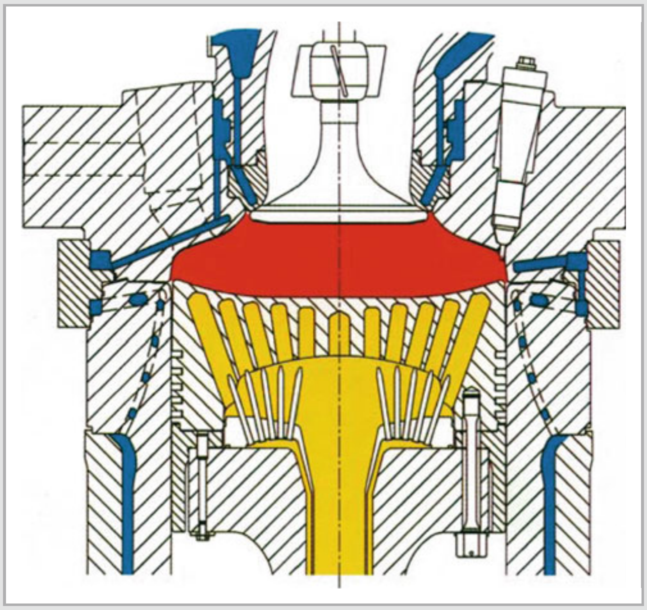

Fig. 18-42
Bore cooling

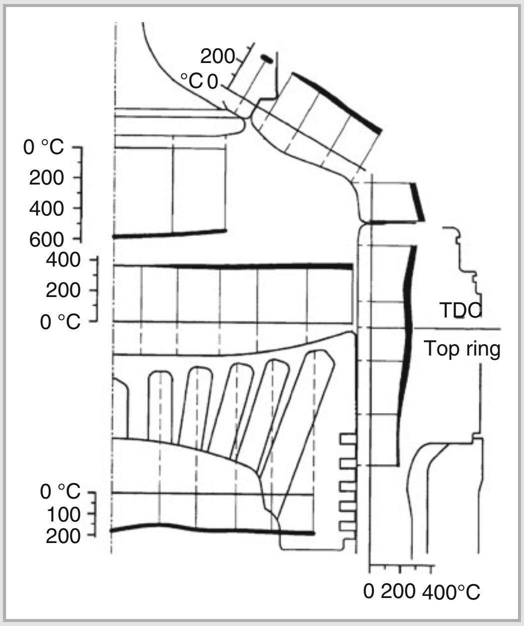

Fig. 18-43 Surface temperatures measured inside the combustion chamber of an 11RTA96C engine with a power of $P_e = 54{,}340$ kW, a brake mean effective pressure of $p_e = 18.2$ bar ($w_e = 1.82$ kJ/dm^3) and a speed of $n = 90$ rpm (R3 power level; see Sect. 18.4.4.1)

Exhaust Valve. Formerly a frequent source of disturbances during heavy fuel oil operation, the exhaust valve has attained remarkable reliability in the engine described here. Eighty thousand hours of operation are reached without maintenance. This can be attributed to the corrosion resistant valve material Nimonic 80A in particular. In addition to the optimal surface temperatures obtained by bore cooling (Fig. 18-43), the valve seat's self-cleaning of combustion residues, effected by the rubbing seating of the rotating valve, also contributes to this. The discharging exhaust gas periodically generates valve rotation by means of an impeller mounted on the valve stem. Hydraulic valve actuation eliminates mechanical vibrations in the valve drive. In addition, the closing force for the valve is generated pneumatically by compressing a pneumatic spring.

Camshaft. The camshaft is driven very precisely by gears, which, unlike a chain drive, also guarantee consistent timing after years of operation. Injection pumps and actuators for the hydraulic valve drive for two cylinders apiece are placed atop the engine A-frame. The engine can be reversed by turning the injection cams with individual servo pumps. The injection pumps in Wärtsilä engines are valve controlled; they are helix controlled in MAN B&W engines. Variable injection timing (VIT) can be applied to optimize part load operation. Injection is performed by three injection valves located symmetrically on the circumference of the cylinder cover. This assures surface temperatures at the piston are optimal (see Fig. 18-43). First implemented by Wärtsilä, the application of common

rail technology is new in large two-stroke diesel engines (see Sect. 18.4.5.2).

Scavenging and Supercharging. A modern two-stroke low speed engine requires a turbocharger with high overall efficiency of up to 72% and a high pressure ratio of up to 4.2. A more compact configuration locates the intercooler near the cylinder jacket. Condensate, which is unavoidable during intercooling, must still be separated before the cylinders to prevent cold corrosion and any disruption of the lubricating oil film.

The symmetrical configuration of the scavenging ports and the exhaust valve facilitates efficient scavenging. The result is a volumetric scavenging efficiency of over 95% compared with values of approximately 85% in loop scavenged engines.

18.4.3 Operating Performance of Two-Stroke Low Speed Engines

The two-stroke low speed diesel engine has earned the reputation of being the most reliable internal combustion engine for good reason. During the last three decades, various factors have induced diesel engine manufacturers to additionally increase the engine's inherent reliability:

- The quality of heavy fuel oil has noticeably diminished since the "oil crisis" in the 1970s because oil refineries have intensified their production of lighter distillate from crude oil by new processes (see Sect. 4.3).
- Despite the demanding operating conditions it entails, customers expect longer overhaul intervals of three years, which corresponds to a ship's usual cycle for periodic maintenance in dry-dock.

Large two-stroke engines have undergone further improvements in recent years to achieve this. Only two of the most important component assemblies, which are decisive for the overhaul intervals, are treated here as examples:

Piston ring and cylinder liner wear. Advances in development in this area are based on:

- fully honed cylinder liners with clearly defined hard phase fractions that optimally distribute the operating load,
- chrome ceramic piston rings that improve running-in performance and operational reliability,
- anti-polishing rings on the upper edge of cylinder liners, which scrape off potential coke deposits on the piston skirt,
- improved lubricating film and reduced mixed friction zones [18-20] by electronically controlled cylinder lubrication systems that precisely meter the quantity of lubricating oil, which is particularly effective for hydrodynamic lubrication between rings and the cylinder liner,
- bore cooling, as already described, for the purpose of optimal component temperatures,
- three injection nozzles for the purpose of optimal mixture formation and combustion temperatures and
- prevention of material attrition on the piston.

This has made overhaul intervals of three years or approximately 18,000 hours of operation possible in present day advanced two-stroke engines. With operating values for a large two-stroke engine measured during propeller operation, Fig. 18-44 documents the conditions under which these intervals have to be produced.

The maximum firing pressure at full load reaches the value of 142 bar and is kept constant between approximately 80% load and full load by the variable timing of the start of injection. Thus, the characteristic of specific fuel consumption as a function of load remains flat. The very low exhaust temperatures (approximately 450°C before and 300°C after the turbine) are also noteworthy. They indicate the two-stroke engine's particularly high efficiency, sufficient energy still being available for the exhaust gas boiler.

18.4.4 Two-Stroke Low Speed Engines as Marine Engines

18.4.4.1 Propulsion System Tuning

Since the diesel engine is part of a system, optimal design of the propulsion system is particularly important. Important parameters are:

- optimal adjustment of the ship–propeller–engine,
- optimization of the auxiliary systems necessary for the engine (lubricating oil, fuel, cooling system, etc.),
- prevention of disturbing or harmful vibrations,
- optimal generation of auxiliary power on board and
- optimal recovery of waste heat.

Optimally adjusting the engine, propeller and ship to one another requires first selecting the diesel engine connected with a fixed propeller from the available models of a series, allowing for the propeller and ship characteristics. Taking the ship's shape and the propeller data as the starting point, the propulsion power can be determined at the selected ship speed on the basis of model tests, simulations and previous examples. The ship's various operating modes, e.g. loaded, ballasted, clean or dirty outer hull, must be considered. Once power and speed are established, the engine is selected. Since the power maps of the individual engines from the series overlap, several engines may often deliver the desired power-speed combination for a particular case. Then, other criteria, e.g. number of cylinders, dimensions, specific consumption, etc., may be referenced for the final decision.

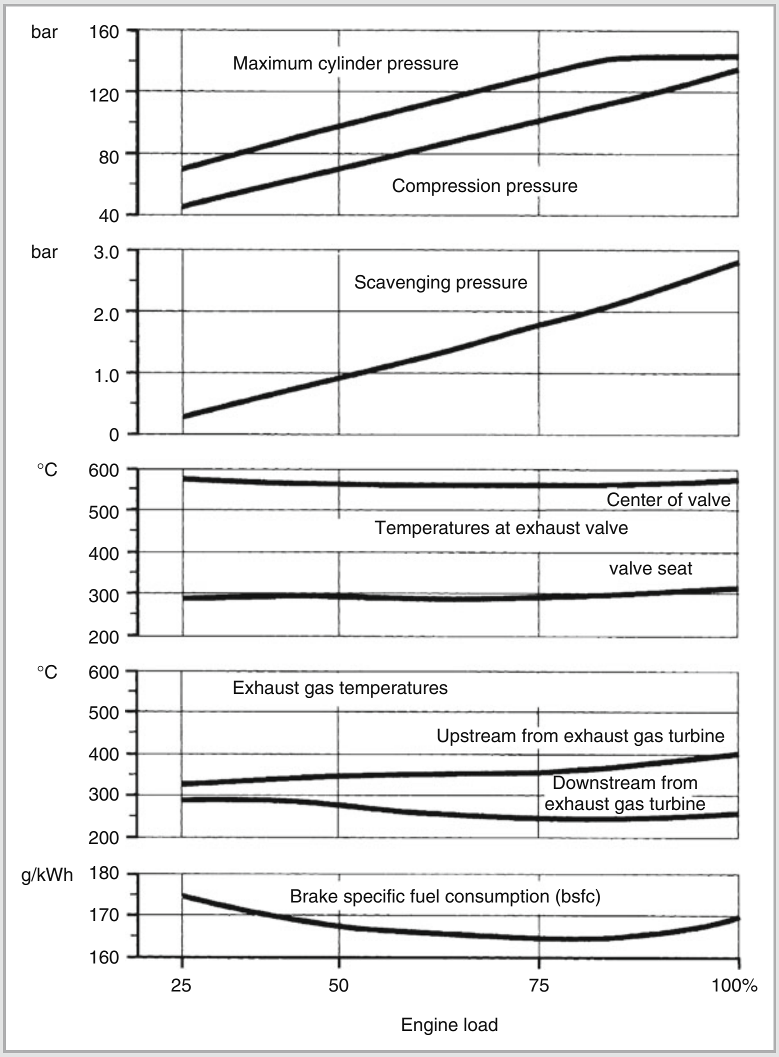

Fig. 18-44
Characteristics of the most important engine parameters of the 11RTA96C engine during propeller operation for R3 power: $P_e = 54{,}340$ kW; $n = 90$ rpm

When the engine is intended to drive other units, e.g. shaft generators, they must be additionally factored in when determining the required rated power. Just like on-board diesel generators, shaft generators can be drawn on to generate the electrical power required on board a ship. The advantage of auxiliary diesel engines is their great operating flexibility. However, the operating costs are somewhat higher when they are not also operated with cheaper heavy fuel oil instead of diesel oil (unifuel concept). By comparison, shaft generators connected with the main engine by gears have lower fuel costs because of the PTO concept (power take-off) but require higher investments in the system. A significant cost factor is the necessary tuning of the power frequency by a gear with a variable transmission ratio or by a thyristor inverter when propeller speed changes.

The low exhaust temperatures of 270 to 300°C at the outlet of an exhaust gas turbine impose narrow limits on the energetically advantageous use of turbo generators.

The high thermal efficiency of a two-stroke low speed diesel engine causes over 50% of the thermal energy to be converted into mechanical work. Thus, less exhaust gas heat is available for recovery than in a four-stroke diesel engine. The exhaust gas heat is primarily utilized to generate steam. The heat extracted from the intercooler's first stage as well as part of the cooling water heat is primarily utilized to

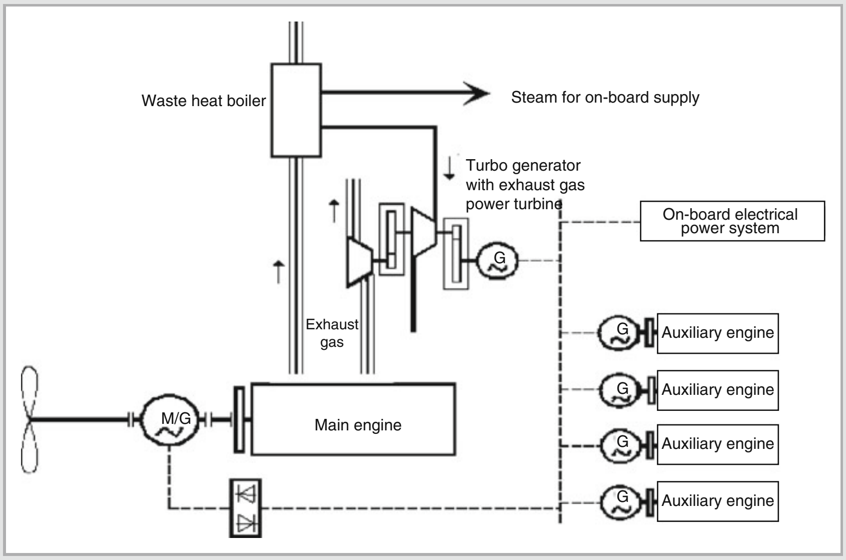

Fig. 18-45
Schematic of a marine propulsion system with a low speed large two-stroke diesel engine and exhaust heat recovery (total heat recovery)

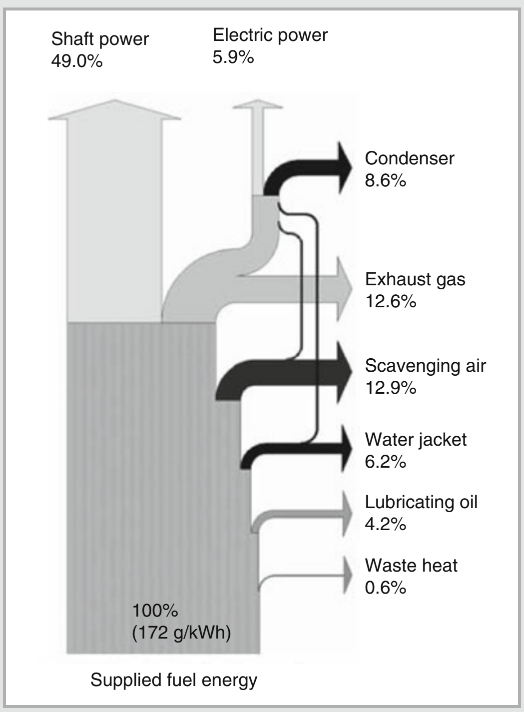

Fig. 18-46 Heat balance of a Wärtsilä 12RT-flex96C two-stroke low speed diesel engine with an overall efficiency of 54.9% by recovering waste heat by means of a turbo generator to generate electrical power

generate warm water or fresh water in freshwater generators (see Chap. 14).

Wärtsilä's expanded turbocompound concept, the waste heat recovery (WHR) system (Fig. 18-45) furnishes an attractive option to recover exhaust heat. The development of highly efficient turbochargers that now have efficiencies of up to 72% makes this concept possible.

After exiting the turbocharger, a majority of the quantity of exhaust gas is fed to a two-stage steam generator that supplies a steam turbine. The turbochargers' high efficiency allows diverting approximately 10% of the quantity of exhaust gas before the turbocharger and supplying it to an exhaust gas turbine, the shaft of which is connected with the shaft of the steam turbine by a gear. The generator driven by this arrangement supplies the ship's onboard electrical system with additional electrical power that a shaft motor can even convert into increased propulsion power for the ship. Thus, in conjunction with the waste heat recovery system, a large two-stroke engine's overall efficiency can be increased to approximately 55% (Fig. 18-46). This reduces CO_2 emissions over a standard engine by approximately 11% and the remaining exhaust emission also decreases in the same ratio relative to the engine power delivered [18-21].

MAN Diesel SE offers a similar concept called the Thermo Efficiency System (TES).

18.4.4.2 Vibration Damping in the Drive Train

The pursuit of higher propulsion efficiency has resulted in low engine speeds for a certain engine power. This was achieved by

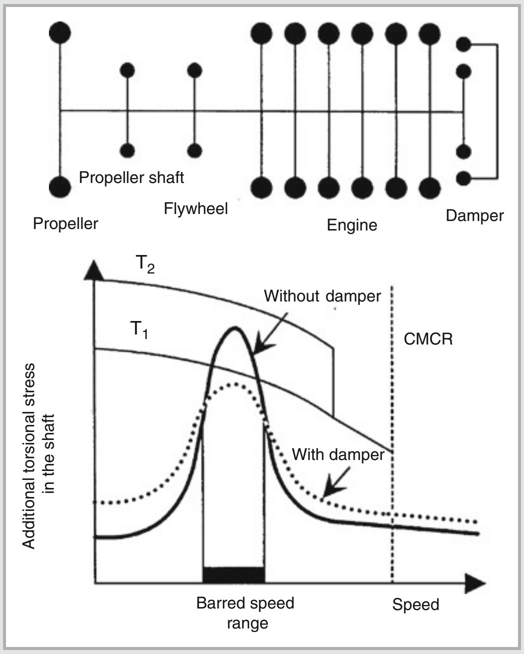

Fig. 18-47 Calculated vibration system of engine–propeller shaft–propeller. Reduction of the torsional vibration amplitude by means of a damper. T_1 is the permissible limit for continuous mode T_2 the permissible limit for transit mode

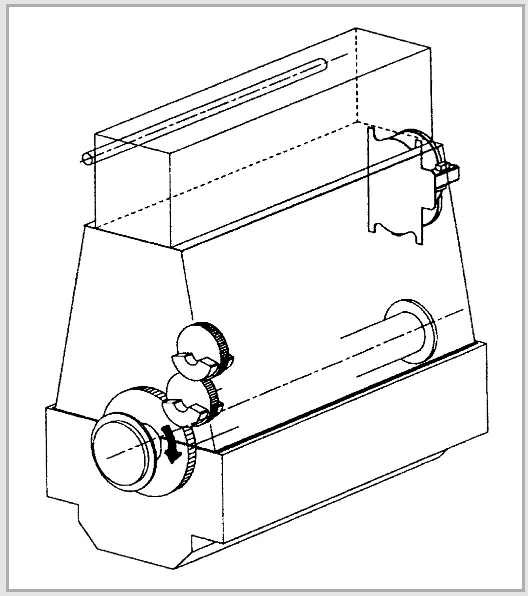

Fig. 18-48 Balancing of the second order moment by electrically driven unbalancers arranged on the free engine end and gear driven unbalancers on the driving end

– increasing the stroke to bore ratio from approximately 2.0 to over 4.0 and
– increasing the use of low speed engines with large bores and consequently lower numbers of cylinders, e.g. four and five cylinder engines.

The trend toward higher s/D ratios has led to a reduction of the natural frequencies of the crankshaft, which is becoming slenderer. Thus, critical engine speeds approach the propulsion engine's operating speed more often, unless this has been prevented by meticulously tuning the vibration system of the engine–propeller shaft–propeller (Fig. 18-47). If fully eliminating the resonances from the operating range is impossible, then the vibration amplitude is reduced by means of a vibration damper. Similar methods are also applied when damping or "detuning" a shaft system's axial vibrations.

The transition to smaller numbers of cylinders requires additional provisions to cancel the unbalanced first and second order free moments in four and five cylinder engines. Optimal placement of the engine in a ship can serve as a corrective: to prevent the excitation of hull vibrations, an engine should not to be placed in a node of the oscillating hull. If the solution is still unsatisfactory, then counterweights on the crankshaft "detune" the first order free moments' phase relation and amplitude. Second order free moments' can usually be balanced by "Lanchester" balancers running at twice the engine speed (see Sects. 8.1 and 8.2). Placing an electrically driven unbalancer on the free end of the engine in combination with the balancing mass integrated in the engine on the drive side and driven by a gear has proven itself for the vertical second order moment $M_{2\ \mathrm{V}}$ that frequently occurs in two-stroke engines with four to six cylinders. Thus, the effect of balancing the moment can be inexpensively tested and adjusted in marine operation independent of load and speed (Fig. 18-48). Mounting an electrically driven second order balancer, which is located as far astern as possible and also covers potentially disturbing effects of the propeller moment, furnishes another option to damp vibrations, especially in ships with bridges located far aft (Fig. 18-49).

18.4.5 Outlook

18.4.5.1 Trends in Future Development

General remarks. The advances in the efficiency and reliability of large two-stroke engines in just the last few decades do not mean that development has already reached its limits. Theoretical analyses by Eberle [18-22] demonstrate the definite possibility of thermal efficiencies above 60% and even higher specific powers. However, since particularly high reliability continues to be required of marine diesel engines,

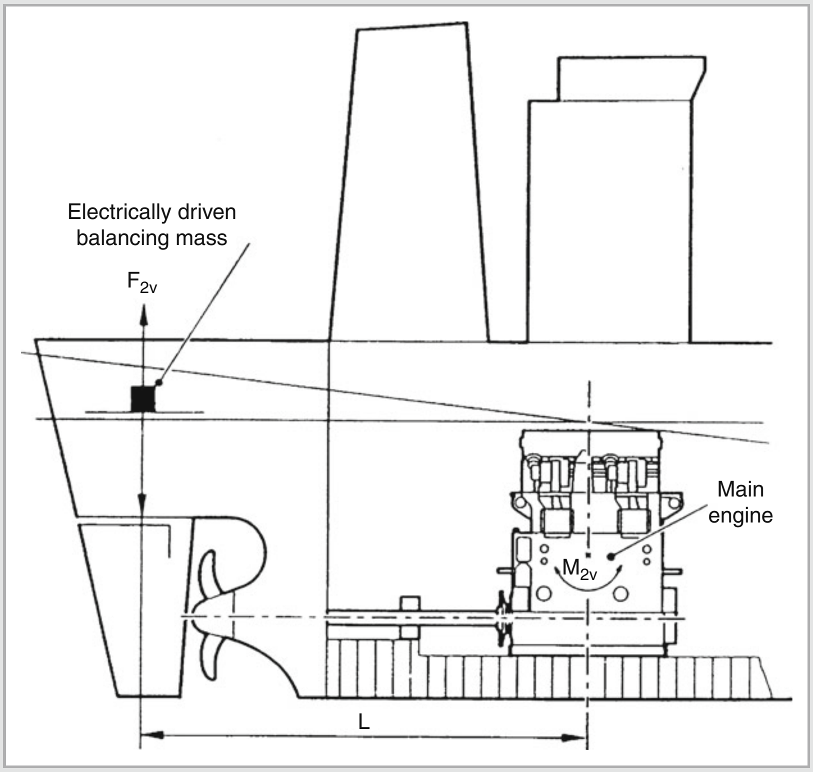

Fig. 18-49
Generation of a free second order force by an electrically driven balancing mass to damp vibrations caused by the moment of inertia M_{2v}

compromises that benefit the power concentration or detract from operational reliability are unfeasible.

The following discusses potential development scenarios for engine characteristics. However, the conclusions contain a level of uncertainty since they may be influenced by unquantifiable constraints, e.g. economic growth, environmental requirements, crude oil availability or technical trends.

Brake mean effective pressure and maximum cylinder pressure. Thermodynamically, a constant ratio of maximum cylinder pressure and brake mean effective pressure (or specific work) corresponds to an approximately constant level of thermal efficiency. Thus, at a proven optimal ratio of $p_{Zmax}/p_e = 8.0$, the maximum cylinder pressure in current large two-stroke engines increases linearly with the brake mean effective pressure when there is no intention to accept any efficiency loss (Fig. 18-50):

- Since high brake mean effective pressures implicitly require high maximum pressures in the cylinder, at a $p_e = 21$ bar, the pressure p_{Zmax} increases to 168 bar.
- Similarly, higher boost pressures of approximately 4.2 bar and correspondingly high turbocharger efficiencies of approximately 72% are essential (see Sect. 2.2).

Such values are now thoroughly realistic. The principle of bore cooling (see Sect. 18.4.2.2) still holds substantial potential for the component temperature level. Tribological development of the mating of the piston ring-cylinder liner will likewise have to follow this trend to maintain the two-stroke diesel engine's traditional reliability.

Stroke to bore ratio and mean piston velocity. When the general definition of engine power P_e (see Sect. 1.2) as a function of the number of cylinders z, mean piston velocity c_m, engine speed n, stroke to bore ratio $s/D = \zeta$ and brake mean effective pressure p_e is examined in a less common formulation

$$P_e \sim z \cdot p_e \cdot c_m^3/(n^2 \cdot \zeta^2),$$

where the power P absorbed by the propeller is a function of design speed $n_p = n$, then the following ensues

$$P_e \sim n^a,$$

where $a = 0.3$ is a function of the stroke to bore ratio of the aforementioned parameters

$$\zeta_2 = \zeta_1 (n_1/n_2)^{1.15} \cdot (p_{e2}/p_{e1})^{0.5} \cdot (c_{m2}/c_{m1})^{1.5} \cdot (z_2/z_1)^{0.5}$$

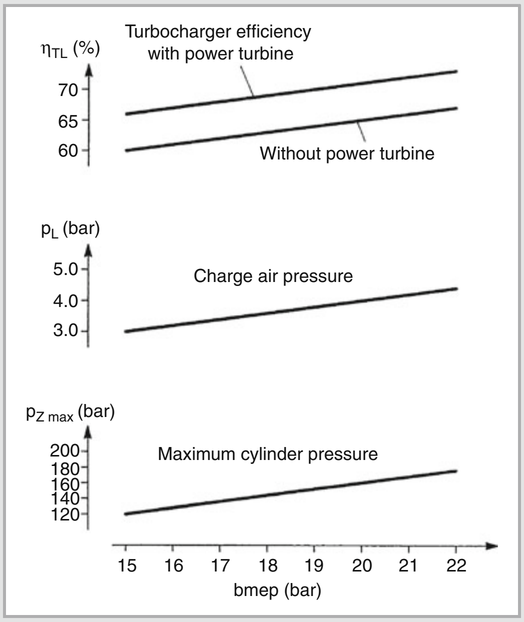

Fig. 18-50 Maximum cylinder pressure p_{Zmax}, boost pressure p_L and turbocharger efficiency η_{TL} as a function of brake mean effective pressure p_e at constant thermal efficiency

The index "1" corresponds to the current "state-of-the-art" reference engine, the index "2" to the potential next stage of development.

The stroke to bore ratio may be expected to increase further

- when the power per unit piston area $P_A \approx p_e \cdot c_m$ is higher and
- even larger, lower speed propellers with higher efficiency are employed.

However, an increase in the stroke to bore ratio is at odds with the higher specific costs and the engine's height and width.

In conjunction with anticipated emission regulations, further increases in the power of highly supercharged two-stroke low speed engines by increasing the brake mean effective pressure to 21 bar and above will make flexible adjustment of the engine with electronically controlled injection and exhaust valve timing imperative. Operating parameters that correspond to an optimal setting at full load will no longer be optimal in the part load range without intervention. In addition, this will necessitate adjustment of the

- injection parameters,
- valve timing and
- cooling and lubrication

as a function of load as well as a high performance supercharging system capable of delivering the necessary pressure ratio in conjunction with the higher efficiencies.

This will allow:

1. optimally adjusting engine performance for consumption, load and emissions at any load and
2. increasing the reliability of components by monitoring and controlling the most important engine functions.

Developed in recent years, common rail injection satisfies the requisite conditions.

18.4.5.2 Common Rail Technology in Large Two-Stroke Diesel Engines

The introduction of the common rail fuel injection system suitable for heavy fuel oil for large two-stroke engines constituted a technological leap in the direction of the aforementioned development. Wärtsilä's RT-flex concept had been tested on a laboratory engine since 1998 and was implemented in a standard engine of the 6RT-flex58T type in 2001. The camshaft, which previously controlled injection and exhaust valve actuation and greatly restricted the potential for optimization, was replaced with electronic timing that furnishes great flexibility to optimize an engine under the widest variety of operating conditions. A central pump unit (supply unit) supplies both heavy fuel oil and hydraulic oil that actuates exhaust valves to the two accumulators in the rail unit located at the level of the cylinder covers (Fig. 18-51). From there, the

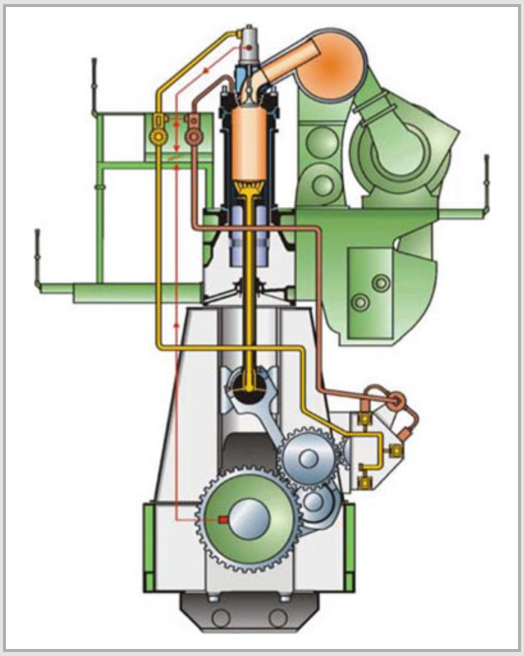

Fig. 18-51 Configuration of a Wärtsilä RT-flex96C two-stroke diesel engine with a common rail injection system suitable for heavy fuel oil and an electronically controlled exhaust valve

fuel and hydraulic oil each travel through a control unit to the three peripheral injection valves or to the central exhaust valve on every cylinder. Fully electronic timing of both functions makes the operating parameters of injection timing and duration as well as the movement of the exhaust valve freely adjustable. While the servo system that actuates the valves operates with a pressure of 200 bar, the maximum injection pressure in the fuel rail is 1,000 bar. A metering piston integrated in the injection control unit controls the volume of the injection for each cylinder. Its position at any time is measured electronically, thus facilitating exact metering of the injected fuel quantity for each cylinder. The three injection nozzles per cylinder are individually controllable and thus also allow cutting off individual nozzles at low loads to improve injection with the smallest quantities of fuel and prevent the formation of smoke at part load [18-23].

The fundamental advantages of the RT-flex system are its reduced fuel consumption and lower exhaust emission. In addition, the technology enables automatically adjusting engine operating parameters to an engine's current state. This constitutes an elementarily important step toward an "intelligent" engine.

Specifically, common rail technology provides the following advantages for the operating performance of large two-stroke engines:

- lower fuel consumption in the middle and upper part load range by variable injection pressure and freely selectable valve timing,
- smokeless operation under all operating conditions,
- precise speed control and stable engine operation even at lowest speeds in the range of 10–15 rpm,
- lower mechanical and thermal load by more uniform combustion and balanced cylinder pressure level,
- easier adjustment of and less maintenance required for common rail components,
- higher operational reliability and availability through integrated monitoring functions and the redundancy of key components,
- lower engine weight (approximately 2 t per cylinder at average bore sizes) and
- lower vibrations.

The starting air system is also electronically controlled and thus allows better engine starting and braking performance than mechanically controlled systems.

MAN Diesel SE also expanded its two-stroke engine program in recent years with an electronically controlled series designated "ME" in which the camshaft has been replaced by an electro-hydraulic servo oil system. Not to be equated with a common rail system, this system utilizes electro-hydraulically actuated single plunger pumps for fuel injection and electronically controlled valve actuators for the exhaust valve function [18-24]. The flexibility of injection pressure, timing and valve timing is comparable to that of common rail systems. Basic differences are the increased requirements for hydraulic damping in the servo oil system, which are satisfied with the aid of pneumatic accumulators, and the elimination of independent activation and cutoff of individual injection nozzles.

18.4.5.3 Exhaust Emissions

The reduction of exhaust emissions is now the top priority in the further development of large two-stroke marine engines. The inherent advantage of a low speed diesel engine is its high efficiency. Carbon dioxide emissions and the fraction of unburned hydrocarbons are very low. This holds true for visible smoke too but not for particulate emission based on the ISO standard, particularly in heavy fuel oil operation (see Sect. 4.3.4.2).

By comparison, the fraction of nitrogen oxides is relatively high compared with other combustion engines with lower efficiency. Regulated by IMO Marpol Annex VI, the nitrogen oxide limit in effect is 17.0 g/kWh for marine engines with a speed below 130 rpm. The following measures are applied in present day large two-stroke engines to comply with the limits:

- higher compression ratios,
- optimized geometry of injection nozzles (number of spray holes, orifice diameter, angle of spray) and
- retarded injection timing.

The potential to further reduce NO_X emission by exhausting all the options listed here remains low (see Fig. 18-52 for optimized NO_X reduction).

A further reduction of the limit for nitrogen oxides in the IMO regulation or local emission control legislation is foreseeable. The following additional options are available to reduce NO_X further (see Fig. 18-52):

- optimizing the injection parameters by common rail injection (CR injection).
- reducing NO_X by between 20 and 50% by engine internal measures employing water (wet technologies), scavenge air humidification, water/fuel emulsions and direct water injection also in combination with common rail injection (Wärtsilä's RT-flex system).
- combining water injection with internal exhaust gas recirculation by shortening the scavenging process (WaCoReG) and thus making it possible to reduce NO_X by up to 70% and
- aftertreating exhaust with an SCR catalyst into which ammonia or a urea solution is injected (see Sect. 15.6) and thus making it possible to convert up to 95% of the nitrogen oxides.

The SCR catalyst in large two-stroke engines has to be placed before the exhaust gas turbocharger since the exhaust temperature level there is high enough to obtain optimal conversion rates and simultaneously prevent corrosion problems. Particularly compact designs allow installation of the catalyst in the exhaust pipe at the engine.

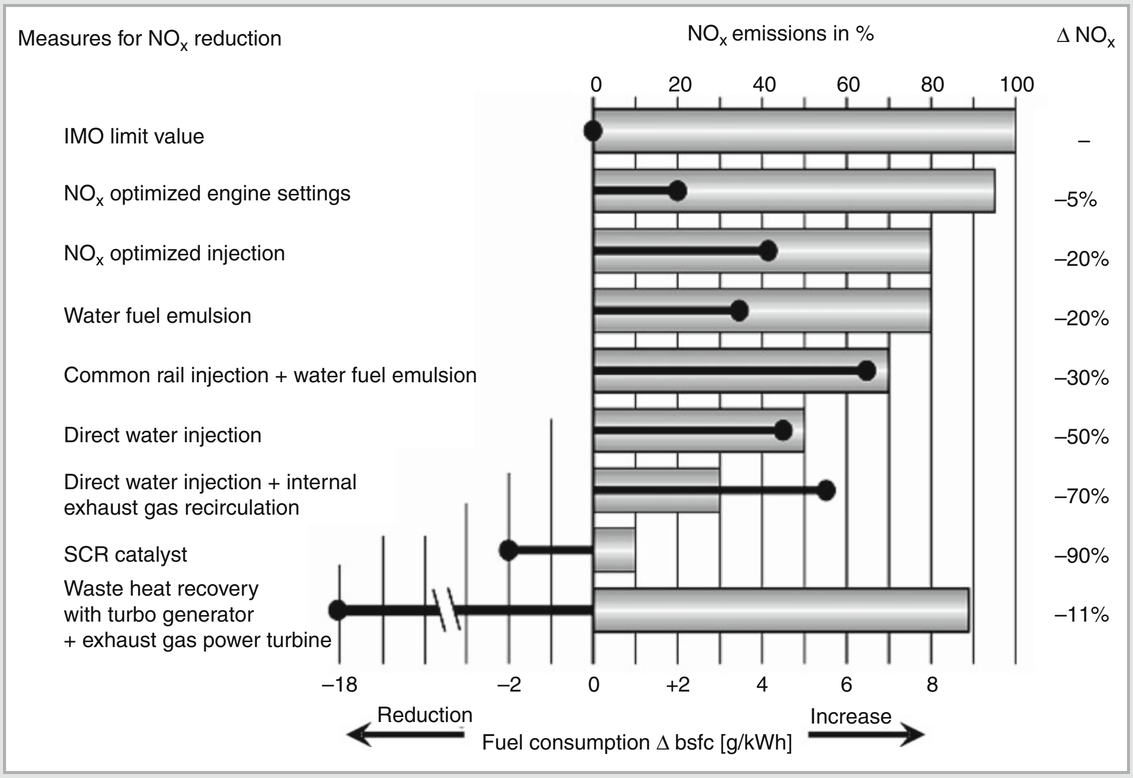

Fig. 18-52 Measures for a two-stroke low speed engine to reduce nitrogen oxides and its influence on fuel consumption b_e, based on the currently effective NO_X limit of 17 g/kWh according to the IMO regulation

Whereas the reduction measures discussed thus far always result in higher fuel consumption (Fig. 18-52), the combustion cycle can be optimized to reduce fuel consumption when an SCR catalyst is used.

Waste heat recovery by means of a steam generator in conjunction with a turbo generator and an exhaust gas power turbine (total heat recovery; see Fig. 18-45) delivers overall drive system power boosted by a total of 11%. This translates into a reduction of nitrogen oxide emission relative to power of the same magnitude and, at the same time, a corresponding increase in efficiency, thus reducing fuel consumption by 18 g/kWh (see Fig. 18-52).

18.4.5.4 Concluding Thoughts

Vying with other combustion engines as the dominant drive source for oceangoing vessels, the two-stroke low speed diesel engine has so far been able to maintain and even consolidate its edge in terms of thermal efficiency, suitability for heavy fuel oil and reliability. Its future prospects are also encouraging since, thanks to technological advances in materials technology, turbocharging and electronically controlled common rail injection, considerable potential still exists in terms of power density, cost effectiveness, reliability and emission performance. The use of state-of-the-art technologies to further reduce exhaust emissions, particularly carbon dioxide, nitrogen oxides, black smoke and particulate matter, will be crucially important [18-25].

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